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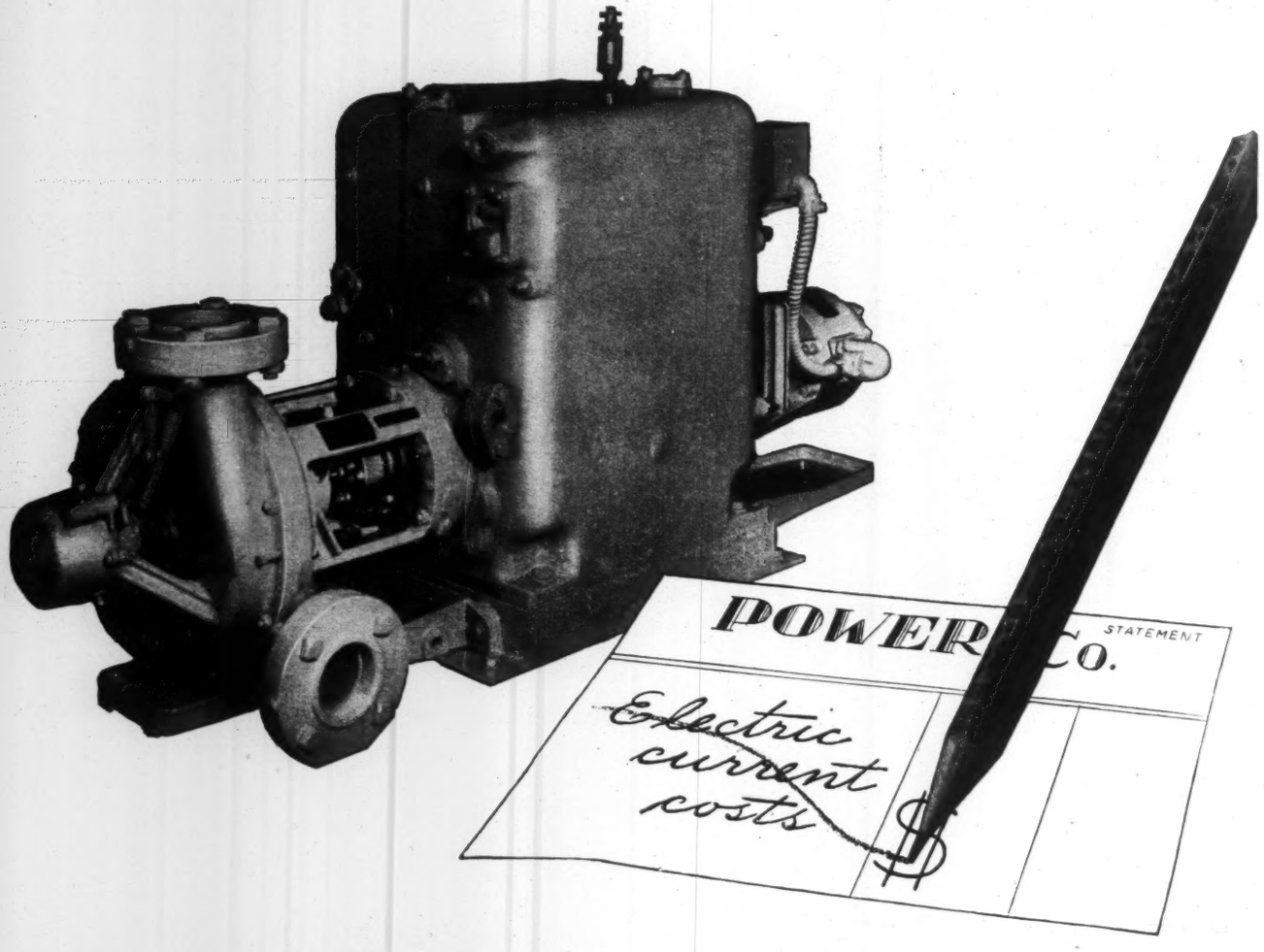

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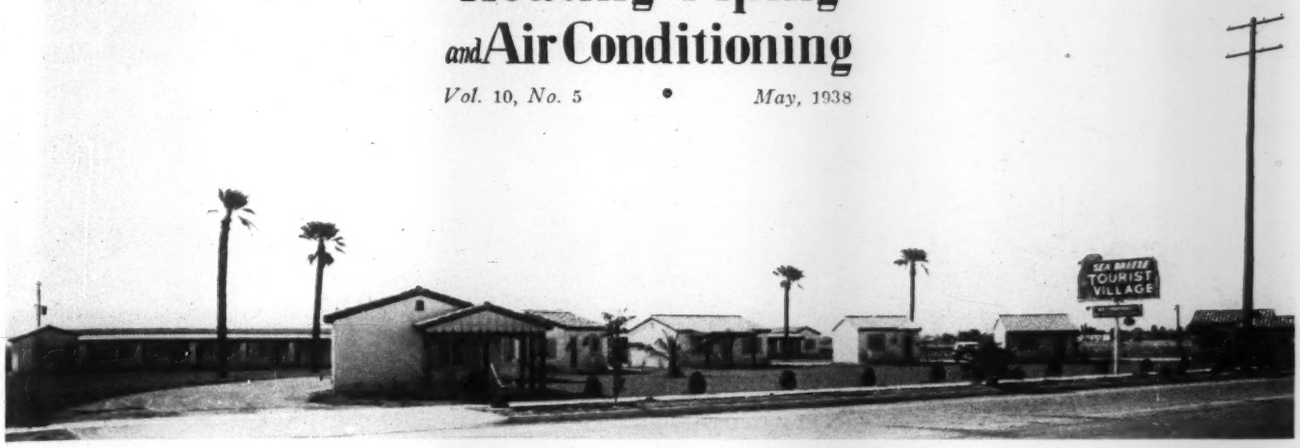


Air Conditioned Tourist Camp–Drive In

By George M. Sebree*

The Sea Breeze tourist village—or "motor court," as they are called in the Southwest—at Phoenix, Arizona, offers a good night's sleep winter or summer, for it has a year 'round air conditioning system. How the air conditioning was designed and installed, what it cost, how it works, and what the public thinks of it are our subjects here. . . . . Mr. Sebree knows intimately what he's writing about, for he is familiar with the installation both from the air conditioning engineer's and the owner's viewpoint. His discussion is frank, complete and authoritative

THE Sea Breeze tourist village, Phoenix, Arizona, is air conditioned, using a refrigerated water circulating system with connections to an overhead finned coil and fan unit in each room. This particular tourist camp—or "motor court," as they are called in the Southwest—was built around its air conditioning. How the system was designed and installed, what it cost, how it works, and what the public thinks of it, are our subjects.

With 15 rooms of a total of 24 ready to rent, the Sea Breeze village opened for business last August. It had—and has—a high investment per room, a good location on the main highway, an experienced manager, and an air conditioning system. Of the 24 rentable rooms, six are individual cottages arranged in a semicircle at the front. These cottages are for the purpose of "dressing up" the place, and to hold to the tourist camp atmosphere and tradition, because the first camps were all detached rooms. At the keystone point of the semicircle is the office and manager's room.

The other 18 rooms are in two blocks of nine each, for economy in building construction and operation of the heating and cooling facilities. One block was not completed until about two months after the court opened for business. Behind these multiple rooms, and on the center of the back property line, is the power house which contains the refrigerating and heating equipment, pumps, etc.

To the Tourist, a Good Night's Sleep

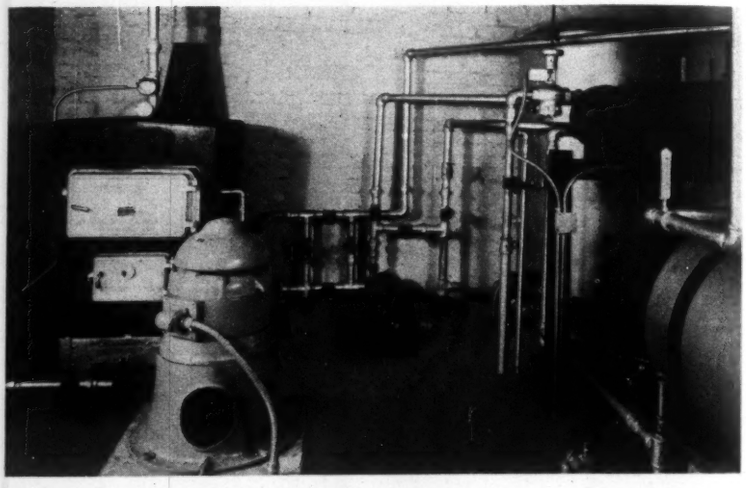
It is hot in Phoenix in August. The Sea Breeze court was ready to open. A neon sign was put out—"Air Conditioned by Refrigeration". To make a good story, it would be nice to say that the court was swamped with business from the moment of opening. As we are dealing with facts, it must be said that four or five rooms were rented the first night. At the end of a week, the nightly occupancy was averaging 10 out of the 15 rooms then available. Before two weeks were up, the court was getting guests who were directed to the place by people who had stopped there. Frequently the guest on registering would remark, "A man I met on the road told me to stop here. He said you had a real air conditioning system".

The court had something the traveling public wanted—real comfort at any season, plus the advantage of rate and convenience that has built the motor court into a big business proposition when viewed collectively. But the comfort part was a little too new for some motor court patrons. They simply couldn't believe it about the air conditioning. It became a necessary—and sometimes difficult—part of the technique of the room clerk to induce the prospective guest to get out of his car and come in the office to discuss rates, and at the same time to sample the cool atmosphere inside. Some of them didn't want to get out of the car. They gave the impression that they were just too tired and too hot to move.

Once in the office, though, or in the room having the

*President, Western Air Conditioning Corp.

Interior of a guest room showing air supply grille above door. Unit is directly behind grille with conditioned air connection to bathroom on left. . . . . *Below*—Apparatus room, showing deep well pump



accommodations desired, there usually wasn't much argument. The air conditioning was functioning and acting like it would continue to function. The guest saw a good night's sleep in prospect and if the rate was mutually agreeable a deal was quickly closed.

Thirty Ton Compressor Cools Water

Chilled water for air conditioning is furnished by a circulating system, the water being cooled in summer by a compressor and shell and tube cooler. The compressor is a two cylinder, 6¼x6¼ in. unit rated at 29.6 tons with 30 F ammonia temperature, driven by a 30 hp motor automatically controlled by the return water temperature. In winter, water in the same lines is heated by an automatic gas fired boiler. Each of the rooms has an overhead finned coil and fan unit, made to fit the space available. The insulated water supply and return lines are buried 3 ft deep.

With a nominal 30 tons available for 26 rooms (including the office and manager's rooms) the court is anything but short on tonnage. The Arizona sun is hot, but the buildings are constructed to withstand it. The individual cottages have tile gabled roofs in variegated colors, mostly good heat absorbers, but above the ceiling is 4 in. of rock wool, and between the ceiling and roof the space is cross ventilated. The multiple rooms have a flat roof with insulation made up of long wood shavings impregnated with concrete. This insulation has a *K* factor comparing favorably with other types of rigid insulation, and was applied in 2 in. thickness. Three ply tar and felt roofing went on top of the insulation and the finished roof was given two coats of aluminum paint.

All walls are constructed of 8 in. concrete blocks in irregular sizes, but flat and painted white outside. Partitions between rooms in the multiple blocks are also of the same thickness blocks. The air conditioning unit is not turned on in any room unless it is occupied, or is next up to be rented, and it was therefore not desired to have too much heat leakage through partition walls. These partitions turned out to be practically soundproof also. Outside walls, partition walls, and ceilings are all plastered with California Mission tinted plaster to an average thickness of ¾ in. All windows have venetian blinds, and doors are 1½ in. solid wood.

Cooling and Heating Calculations

Calculations for cooling were based on maintaining 77 F inside when the outside temperature is 125 F. "A difference of 48 F," you say, "enough to kill you". The answer is that this, or more, is the difference that can be and is maintained regularly in many a Phoenix home, office, store, hotel and club. Furthermore, no one seems to suffer any bad effects. Midsummer heat in Phoenix is withering but not wilting. Going from a conditioned space into outdoors does not usually expose one to the jolting impact of heat plus humidity.

For heating, the system was figured to maintain 72 F inside when the outside temperature is 35 F. This is a 37 F difference, and except in very exceptional cases will occur only at night. The hot water boiler is oversize however, as is the cooling equipment, to take care of some rooms which may be built later. Approximately half the boiler capacity is reserved for heating domestic hot water, so that in case of a phenomenally rapid fall in outside temperature the domestic hot water heater could be cut out temporarily and hot water for the bathrooms would still be supplied for three hours by a 500 gal insulated storage tank. This would make double the normal boiler capacity available to cope with the situation. But during the winter of 1937-38, no condition arose to give the heating capacity of the system even a good workout.

Why Not an Evaporative System?

By now some of my readers will be asking "Why wasn't an evaporative system used, with its big saving

in installation and operating cost over the refrigerating system?" It's a fair question, too. Phoenix is an air conditioned city. Its skyline is topped by innumerable cooling towers, at the rate of about one cooling tower for each two business establishments. Most of them are connected to an evaporative air cooling system. And on many of the days when the thermometer reads the highest, these evaporative systems are doing a very acceptable job.

Still, it was felt that refrigeration was a necessity. The points against the evaporative system seemed too strong. There is a saying in Phoenix that "When there's a cloud in the sky, these evaporative systems fold up like a tent". The country immediately surrounding Phoenix is not desert; it is irrigated farming country, and irrigation is growing rapidly. The old timers say that the relative humidity has increased appreciably since irrigation was started, and will continue to increase as the water is brought to more and more land.

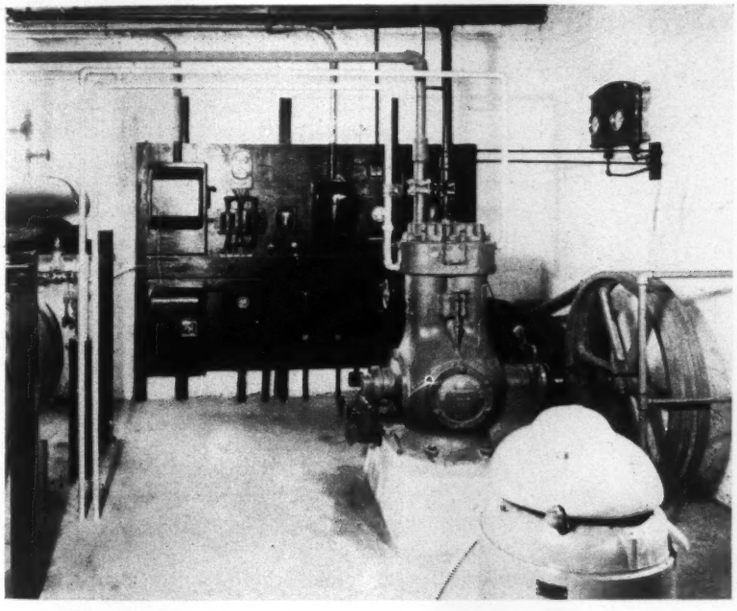
Another point was that an evaporative system would have unbalanced the installation by making necessary a much larger coil for cooling than would have been required for heating. Use water not quite so hot—say 110 F instead of 160 F? Sure—and then run into trouble keeping the domestic hot water hot enough, or to avoid this go to the expense of another boiler or battery of storage hot water heaters, for heating the domestic service hot water only.

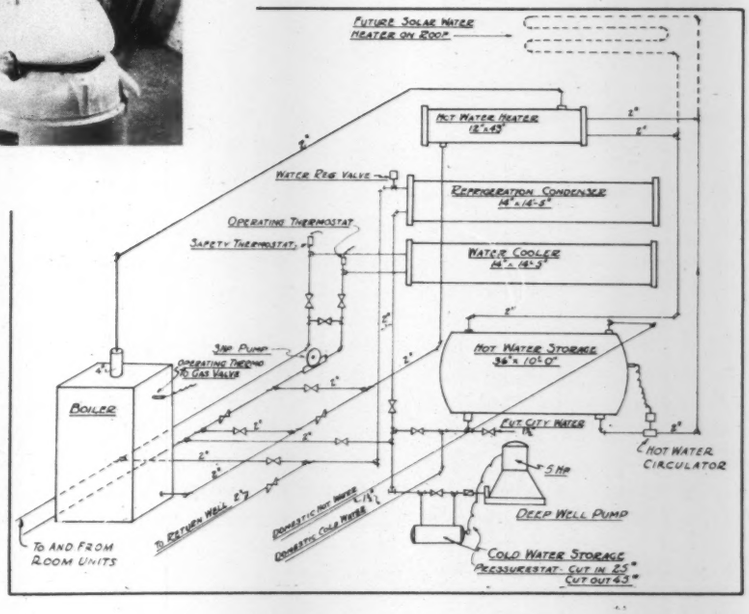
Cost of the System

Finally, the additional cost of the refrigerating system wasn't as much as might be expected. The entire system as installed cost about $12,000. With the connections to the 26 rooms, this comes to about $462 per room. Taking into consideration the probable full capacity of the equipment at 36 rooms, adding $175 per room for 10 additional rooms to cover the cost of coil and fan units and pipe connections, we arrive at a final cost of approximately $382 per room for the system as it will be in the future.

High enough. Still, not so high considering that everything is included in this cost—the refrigerating equipment and power wiring, the boiler, gas burner, and gas connections, two deep wells and a deep well pump, the circulating pump, the hot water storage tank, the hot water heat exchanger, the coil and fan units in the rooms, the water and refrigerant connections, and hundreds of feet of buried and insulated water circulating lines, and the labor to install all of this equipment. If an evaporative system had been used, $3,500 for refrigerating equipment would have been saved. With that deduction in mind, let's start adding the cost of larger underground water lines and larger coil and fan units. This would have resulted in an additional problem because the present units take up all the ceiling space that can be spared in the rooms. Finally, let's add the cost of a good sized cooling tower—not needed now because the deep well pump is necessary anyhow, and it supplies plenty of water for the condenser.

It will be seen that the additional cost (if any) of the refrigeration system over the evaporative was pretty reasonable. Operating cost might be another story, of course. Here it was felt that it was worth something to have the desired conditions on tap any day and every day—to advertise comfort and deliver it





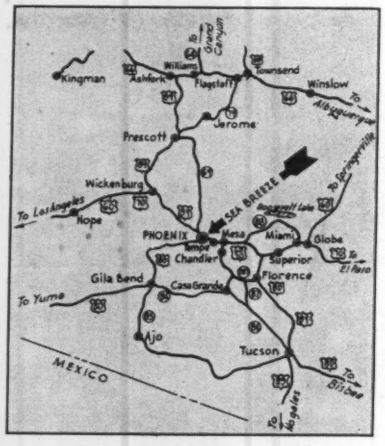
Another view of apparatus room, showing the 30 ton ammonia compressor which chills water circulated to the room units through underground piping. . . . . *Below*—Schematic piping diagram, showing valves for following cycles of operation: (1) Heating domestic hot water by thermo-siphon circulation of boiler water with simultaneous forced circulation of chilled water. (2) Forced circulation of hot water for heating rooms, through boiler, then through shell and tube domestic water heater, with water cooler out of circuit. (3) Connection for circulating well water through the room conditioning units direct in case of the complete breakdown of the refrigerating plant

as advertised and strictly according to specifications.

The Ammonia Refrigeration Plant

The refrigerant selected was ammonia because, in this particular case, there were no disadvantages in using it. There is 60 ft of wide open space between the power house and the nearest guest room. The only duty of the refrigerating equipment is chilling water, which in turn circulates in a closed circuit. All matters of safety are thus taken care of. The water is cooled in a shell and tube cooler having 260 sq ft of surface; about 8.7 sq ft per ton at a nominal capacity of 30 tons. The condenser is almost a duplicate of the cooler, having also 260 sq ft of surface. The refrigerant charge is 300 lb of ammonia which fills the cooler shell about two-thirds full. Both the condenser and water cooler are code welded.

Refrigerant control is by means of a high pressure float valve; consequently 98 per cent of the refrigerant is in the cooler at all times. The compressor is operated by a thermostat in the return water line from the room units. When using only 17 of these room units, the thermostat was set at 42 F cut out, and 50 F cut in, outgoing water temperatures. The cycle of operation was about 8 minutes on and 12 minutes off—three full cycles in an hour. This produced the desired temperature of about 78 F inside the rooms with the hottest outside conditions. As autumn approached and the daily outside maximum dropped to 105 F, then to 100 F, the adjustable differential thermostat was reset to cut out at 44 F and in at 58 F. The running period stayed at 8 minutes, but the off period lengthened to 20 minutes. About October 15, the refrigeration was shut off for the season, and something like 30 days later a little heat began to be needed in the rooms at night.



Water for the condenser is obtained from a 6 in. diameter well 70 ft deep. In summer, the well water temperature is 72 F. With 8.7 sq ft of condensing surface per ton, the condensing pressure averages 160 lb, being a little higher at the start of the 8 minute running period and a little lower at the end. The court is beyond the city water mains at present, and the well is the sole source of supply for all water except drinking water. There is probably nothing wrong with it for drinking too. It has a mineral taste, but is not unpleasant, and chemical and bacteriological tests have shown nothing harmful in it. However, for the present, bottled water with a water cooler is supplied for all rooms.

The well has produced 90 gpm on sustained tests. The water not used for domestic purposes, including all the condenser water, is put back in the ground through an 8 in. return well about 60 ft from the supply well. A rather surprising thing is that it has turned out to be cheaper, more practical and convenient, to use the irrigation district's water for the lawn than to use the well water, which is already on tap under pressure, and most of which is being returned to the 50 ft level underground.

Boiler Is Gas Fired

The boiler is made of steel, insulated, and has a nominal hot water rating of 1100 sq ft EDR. Natural gas is the fuel, and the burner is automatically controlled by the water temperature, being usually set to cut in at 140 F and out at 180 F. Domestic hot water is heated by the boiler water at all seasons of the year by means of a horizontal shell and tube heat exchanger having 45 sq ft of surface. In winter, this heat exchanger is in the circulating pump circuit and in summer the boiler water circulates through the heat exchanger by thermo-siphon action.

All the usual methods of heating water were considered, and they all had one thing in common—the heater itself was sure to lime up quickly with the available water. The shell and tube exchanger was selected because it is a simple matter to take off the heads and clean out the tubes with a suitable cleaner. This exchanger has four passes and the water to be heated is forced through it by a ¼ hp circulating pump operated from a thermostat in the hot water storage tank.

Some of my readers who are still sticking with me will ask, "What's the matter with a solar water heater. Isn't this a natural setup for it?" It's true that a solar water heater could be used with good results in summer and probably will be used at some future time. It is not being used now because the gas utility's connection or minimum charge will probably cover the cost of fuel used. If this cost materially exceeds the minimum charge a solar water heater will be installed because the sunshine is going to be there, right on the roof of the power house, and only a few feet from the water connections to the heat exchanger.

Two Sizes of Room Units

The coil and fan units used in the rooms are of the overhead type, pulled up tight against the ceiling. A natural location suggested itself in the little hallway at the rear of each room. The unit is concealed by a plywood panel underneath and the return air grille is in this panel. Outside air is taken through an 8 in x 6 in. opening in the outside wall, with a louver and screen, near the entering air end of the unit. A short length of discharge duct work is used, most of the conditioned air being discharged straight ahead into the room with a little being deflected through a smaller grille into the bathroom.

All the rooms are not of the same size, the larger single cottages and end rooms in the multiple blocks requiring considerably more capacity than the interior rooms in these blocks. This difference was compensated for by using coils six rows of tubes deep in the units for the larger rooms, four rows of tubes deep in the others. All coils are of the same face area, 9 in. x 16 in. All fans in the units are 6 in., double inlet, double width, and are driven by belted 1/6 hp motors at approximately 1250

Air view of the Sea Breeze tourist village. The building at the rear houses the air conditioning machinery (right half) and linen storage (left half)

rpm in four row coil units, and 1350 rpm in six row coil units. Each unit delivers about 350 cfm, and the difference in capacity and delivery air temperatures between the four and six row coils seems to compensate perfectly for the difference in the size and exposure of the various rooms.

With the return air opening 7 ft above the floor, some trouble with stratification was experienced when heating. This trouble had been expected and was taken care of by using small deflectors over the grilles to force the warm air to the floor. It would be a better solution to take return air from the floor, but space simply does not permit. The fans make some noise, but not enough to be objectionable or to justify lining the inside of the discharge ducts with sound absorbing material. The fan motors are turned on and off by a convenient wall switch marked "Air Conditioning". Individual thermostats have been considered to operate the fan motors and they may be installed some day.

The Piping

The underground water supply and return lines are buried 3 ft deep, and are insulated with wool felt 1½ in. thick. After the pipe covering was applied it was painted with hot asphalt, mopped on with a paint brush. The piping was sized for a total head of 54 ft, and 70 gpm is circulated through it by a 3 hp single suction centrifugal pump. From the power house wall to the farthest unit is a distance of 195 ft along the pipe lines.

The drains from the unit drip pans are not trapped into the sewers. The water in the traps evaporates immediately when the summer cycle is ended. Then, unless a complicated system of double trapping were used, the sewer gas would soon be where it would do the most harm—right in the air stream. A better solution, and the one used in this case, is to run each individual drain line 2 or 3 ft outside the building, about a foot below the ground surface. These lines terminate there with a strainer, in a hole filled with coarse crushed rock, and with the rock covered over with dirt; the water is thus readily absorbed into the surrounding earth.

Such is the picture of a completely air conditioned tourist camp. From a mechanical standpoint it works. From an investment standpoint, what is the value of the air conditioning system? Will it keep the court full during the summer months, when it would otherwise be empty? Will it do this at an average rate satisfactory to both the owners and the traveling public? There is no precedent to go by, but by all signs the answer should be "Yes." This summer will give the answer definitely, one way or the other.

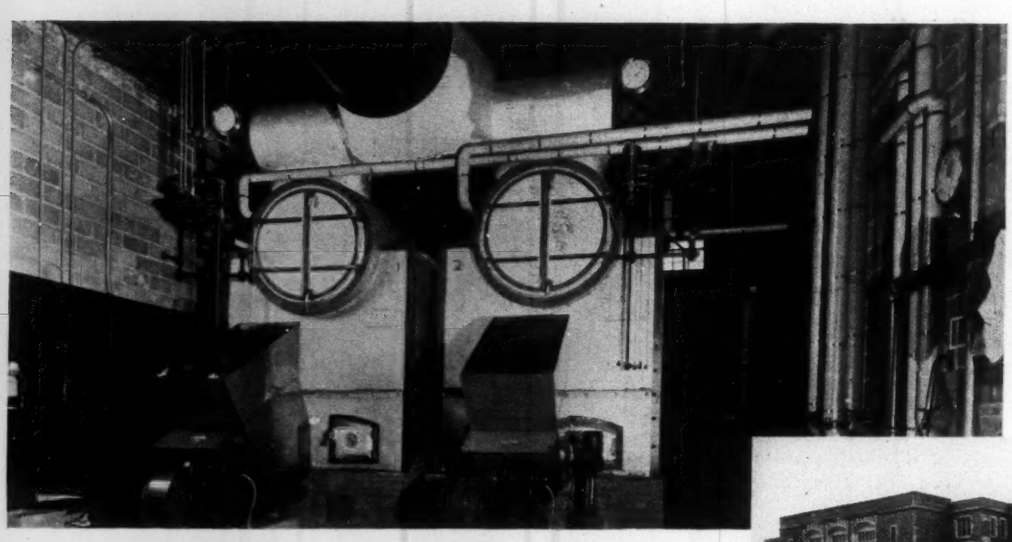
Building Conditioned for $1.00 per Square Foot

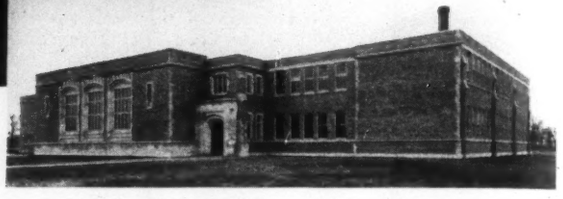
Total net rentable area in the John Mariner Building, Milwaukee, Wis., above the first floor is 40,000 sq ft. This area was air conditioned at a net cost of approximately $1.00 per sq ft. The low figure is undoubtedly due to the fact that the building was designed for air conditioning, which means that the tonnage was low per unit of space and that the equipment was installed as the building construction proceeded.

The management anticipates 100 per cent occupancy of the building by December 1. The rentals asked are approximately 25 to 30 per cent higher than rentals for space in buildings similarly located, but without the convenience and advantages offered by this building. The building management attributes about 80 per cent of this higher rental to the added value offered by air conditioning. This, together with similar experience in other cities, tends to indicate that air conditioning installed in new office buildings is sound from the economic standpoint, that it will not only carry its part of the load but will greatly improve the earning power of any commercial office building.

The upper five floors are served from a central station air conditioning system, with the compressor in the basement engine room, and the air circulating equipment in the penthouse above the sixth floor. The basement equipment consists of one 10″ x 7½″ vertical single acting "Freon-12" compressor, with partial bypass, V-belted to a 60 hp, 208 volt, a-c motor with special starter; one 8″ x 6″ vertical single acting "Freon-12" compressor, with partial bypass, V-belted to a 30 hp, 208 volt a-c motor with special starter; and a 12″ x 10′ horizontal shell and tube condenser.—T. M. Hughey.*

*Sales Engineer, York Ice Machinery Corp.





Left—**Two stoker fired boilers furnish steam at a maximum pressure of 75 lb for heating the school . . .** *Below*—**Exterior view of the building, which has complete facilities for 450 students from first grade through senior high school. The boiler room is at the right rear**

High Pressure Steam Heats Consolidated School

THE Consolidated School of Pleasant Township, completed last year at Kouts, Ind., is heated by a system designed for operation at a maximum pressure of 75 lb per sq in. Architect Callix E. Miller, A.I.A., of South Bend, Ind., adopted the high pressure design for reasons of economy, as his preliminary studies indicated that the savings from small pipe sizes, less required heat emitting surface, elimination of pressure reducing valves, etc., would be much greater than any increase in cost due to the use of high pressure rather than low pressure equipment.

By "consolidated" school is meant one that serves several country school districts or townships. Through pooling the money which would otherwise be spent for several small buildings at various locations, it is possible to erect a large building with complete facilities, comparable to a city school, for children in rural areas. Buses transport the children to and from their homes.

The building is ample to accommodate a student body of 450, has facilities for complete grade, junior and senior high school work, and is provided with complete equipment, including a gymnasium, a separate auditorium with a seating capacity of 460, and well furnished class rooms, shop, laboratories, etc. The cubage is about 700,000 cu ft and the total cost was $176,935.72, which included all furnishings and equipment, but not the architect's fee. The cost of the heating and ventilating systems, including the pneumatic temperature control, boilers and stokers, totaled $19,894.00. On a cubic foot basis, total cost was 25.3c per cu ft and heating cost was 2.9c per cu ft. One of the photographs shows an exterior view of the building, indicating the type of architecture and construction.

For comparative purposes, Mr. Miller cites a school building erected during the same period in another Indiana township, which is heated by low pressure steam. This building comprises an elementary school only, is of a colonial type of architecture, and has a cubage of about 580,000 cu ft. Total construction cost was $191,400 (not including architect's fee) and the cost of the heating and ventilating equipment was about $31,900. On a cubic foot basis, total cost was 33c per cu ft and heating cost was 5.5c per cu ft.

The new Consolidated School of Pleasant Township at Kouts, Ind., is heated by a high pressure steam system designed for a maximum pressure of 75 lb per sq in., with arrangements for varying the pressure depending upon the outside temperature. Primary reason for using high pressure steam was the expected economy in the cost of installation due to smaller required pipe sizes, less radiation surface for the same amount of heat, etc. Steam is used for no other purpose than heating. . . . The following description of this unusual school job is based upon the specifications and other information furnished through the courtesy of the architect, Callix E. Miller

Boilers Are Stoker Fired

Steam for heating at the Kouts school is supplied by two stoker fired, fire tube boilers of 80 rated horsepower each, built to comply with requirements for 100 lb working steam pressure. There is no other use for steam except for heating. The stokers are guaranteed to carry a load of 15,000 sq ft of equivalent direct radiation on each boiler when burning a free burning bituminous coal of not less than 12,000 Btu per lb. Stoker controls consist of a combi-

nation fuel cutout and pressure control instrument on each boiler set for a 75 lb maximum working pressure and a master control instrument having a range from 0 to 100 lb. Magnetic thermal overload cutouts and a hold fire control of the electric time switch type, operating on a three hour cycle with adjustable time cam, are provided. The boiler safety valves are set at 100 lb.

By means of the master control instrument it is possible manually to vary the pressure of the steam supplied by the boilers to the heating system over a range from 0 to 75 lb depending on the outside temperature. During the 1937-38 heating season, which was the first full year of operation, the highest pressure that was needed was about 45 lb on a day when the outside temperature was slightly below zero. The system circulated freely to the most remote radiators on a test at 5 lb boiler pressure during normal mid-winter weather.

The boiler room is at the right rear corner of the building (see photograph) and steam is conveyed through pipe tunnels principally under the first floor corridors to the risers supplying the direct and indirect radiation. The first floor of the building primarily is of concrete slab construction resting directly on the ground, excepting for the plenum and pipe tunnels.

The Radiation

The direct radiation comprises two different types, both standard units of a steel fin type, embodying a heavy copper tube arranged in the form of a continuous coil to which is mechanically bonded a series of steel plates or fins, the whole assembly being rigidly locked into a single unit. Designed for a working pressure of 150 lb per sq in., the radiators were tested in the process of manufacture under a hydrostatic pressure of 5000 lb. Construction of the radiators and the design of the fins is such that there is a high rate of air flow over them, which reduces the amount of radiant heat and tends to keep the surface temperature of the radiators low.

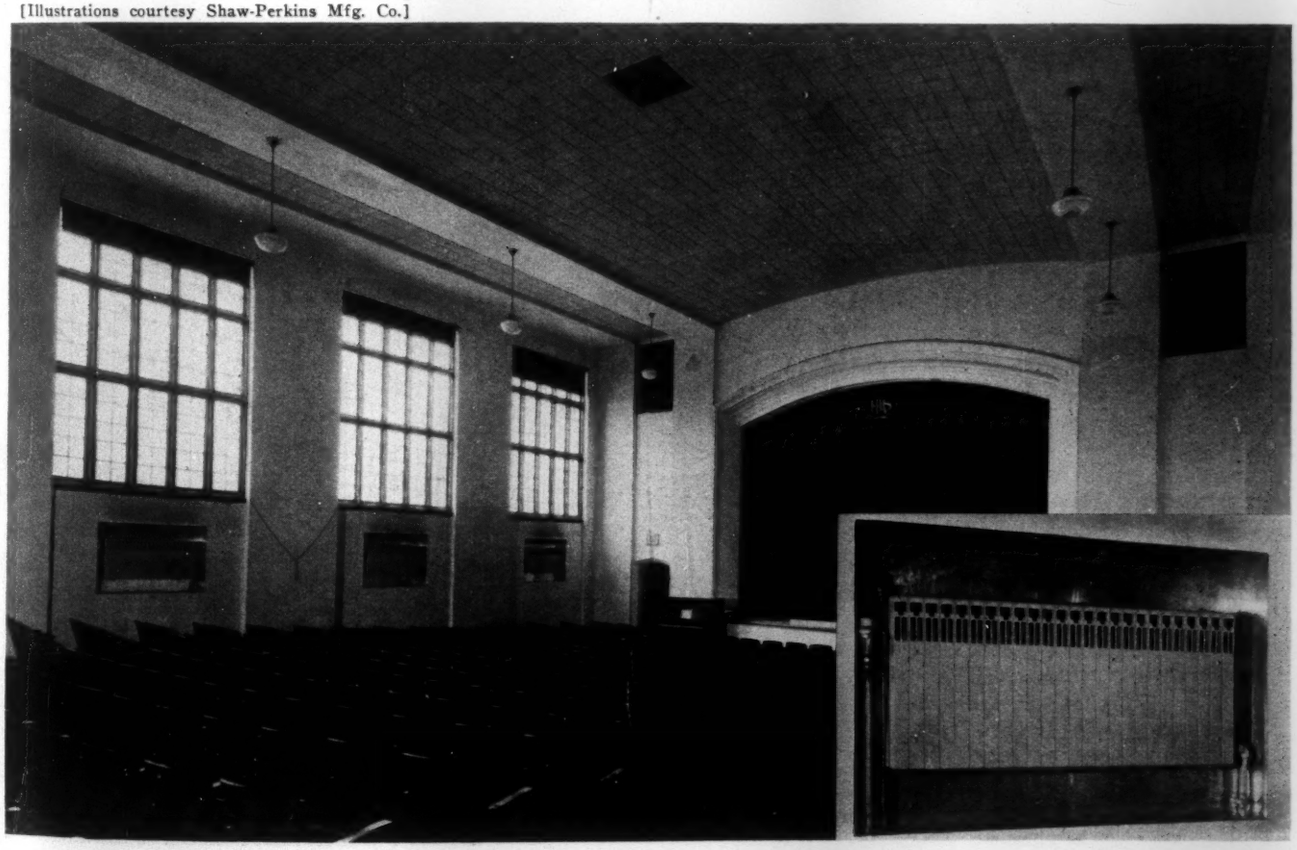
According to the architect, the advantages of using the high pressure steam for heating come from the small radiator sizes for a given amount of heat, the small steam piping and the low cost of insulating this piping. He roughly estimates the pipe sizes were reduced about one-half, and that in general one radiator instead of two was required in all of the class rooms except the commercial room, which cut the number of valves and traps needed. In all, there is 4809.8 sq ft of equivalent direct radiation for the direct radiators, and 21,670 sq ft of equivalent direct radiation for the domestic hot water heating, blast coils, and the unit heaters which serve the gymnasium. The largest steam line in the building is 3½ in. diameter, except for a 4 in. line to the hot water storage heater and the blast coils in the fan room.

The piping was carefully sized to balance steam distribution. With a maximum pressure of 75 lb available for extremely severe weather, heat control is possible by lowering the pressure used in milder weather.

All radiator traps were specified to be good for 100 lb per sq in. working pressure.

The school auditorium, with a seating capacity of 460, showing the three direct radiator units which care for the heat losses along the exterior wall. The close-up shows one of these units, each of which is made up of two radiators, one with front outlet and one with top outlet. The enclosure is backed with reflective insulation

[Illustrations courtesy Shaw-Perkins Mfg. Co.]



The commercial department class room, which is the farthest from the boiler room. This is the only class room with two radiators, all the others having but one radiator

The direct radiation cares for the heat loss from the building and ventilating air is tempered by blast units supplying the auditorium, gymnasium, laboratories, shop, and all class or recitation rooms. These units are designed for a working pressure of 150 lb, but are supplied with steam at a maximum of 75 lb pressure. The indirect heating units have a pan type humidifier with float box and float valve to maintain a constant water level, installed between the tempering heaters and the supply fan. Submerged in the pan is a closed heater of 2 in. red brass pipe to evaporate sufficient moisture to maintain the air supplied for ventilation at a relative humidity of 50 per cent when the entering air is at a temperature of zero. The humidifier is controlled automatically by a gradual acting insertion humidistat in the fan discharge.

The gymnasium outside air supply is tempered by two unit heaters in the regular air supply flues from the plenum chamber to the gymnasium; provisions for recirculation permit quick heating up.

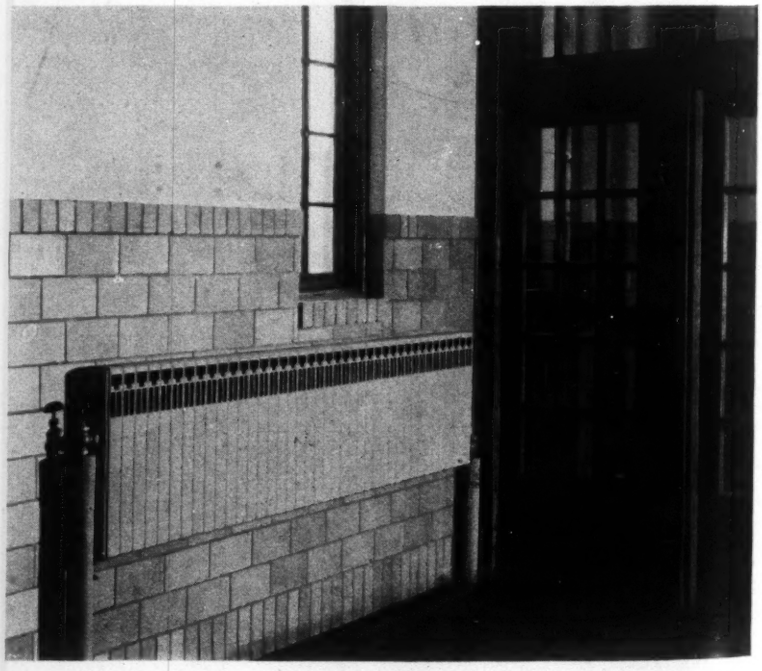
One of the illustrations shows a general view of the 460 seat auditorium with the three direct radiator units that care for the heat loss from the room along the outside wall. The inserted illustration shows a close-up of one of the radiator units in its enclosure, which is backed by reflective foil insulation. These radiators are to be covered with ornamental grilles but were installed without the grilles for the time being because of a limited appropriation. It will be noted that each of these three units is made up of two radiators, one arranged for front outlet of the heated air, with one for top outlet of the heated air immediately behind it. In this way a large amount of heating capacity was gotten into a comparatively small enclosure.

Another of the illustrations shows a general view of the commercial department—which is the farthest room from the boilers—and which is the only class room that has two radiators, all the others having one. Another of the photographs shows a radiator at one of the entrances to the building, and indicates the neat appearance of the installation.

At present, only the blast heating system, the radiation for the gymnasium, and the radiation for the auditorium are under automatic pneumatic temperature control. However, the valves for all radiators have special bodies and can be converted to automatic control by removing the hand control elements when the appropriation for this additional work becomes available later. All roughing-in work is now installed.

The heating contractor for this building was the Valparaiso Plumbing Co., Valparaiso, Ind.

Drs. C. Sleeth and E. Van Liere reported in the *American Journal of Physiology* (Vol. 118, No. 2) that in a cool atmosphere food was digested better and quicker due to an increase in gastric activity. Their report was the result of a study conducted with dogs.



Close-up of radiator at one of the building entrances, indicating neat appearance of installation

Maintenance Routine for a Dust Control System

By Donald E. Frazier*

The use of dust arrester equipment has been stimulated by the ever increasing legislation intended to improve plant working conditions, but without the proper care and maintenance expensive dust collecting systems fail to achieve their purpose. . . . The air cleanliness throughout the mill is a yardstick by which the arrester efficiency is best evaluated. This may be determined by a number of instruments, the most generally accepted being the Smith-Greenburg impinger. Assuming the arrester installation in question is adequate the best insurance against failure is frequent check-ups, which should determine the condition and resistance of the filters, the condition of the piping, air flows, and general effectiveness of the system by means of the dust count. Replacement of any worn parts should follow such a check-up

USE ot arrester equipment for collection of injurious vapors and dusts is being stimulated by the ever increasing legislation intended to improve working conditions. Much such legislation is of a conflicting nature which tends to defeat its purpose. Some states have established codes of specifications which are very drastic, but even so are hardly a guarantee that dangerous conditions will be eliminated. In order to be in accord with social legislation many industrial concerns—especially those handling silica dusts—have installed expensive equipment for removing these dusts from the air; a number of these installations fail in achieving their purpose for lack of proper care and attention. It is hoped that this description of maintenance routine followed at the Bird & Son, inc., crushing plant, East Walpole, Mass., will be of value to those interested in dust arrester installation and maintenance.

This plant is designed to crush small aggregates of approximately 10 to 28 mesh sizes at a rate of 5 tons per hr. The rock as quarried is brought to the mill and the initial crushing is performed by two gyratory crushers. After a preliminary grading on two totally enclosed screens the material goes to a roll mill where it is crushed to approximately the correct size. It then passes through an air separator to remove the fines and then it goes to a large vibrating screen, where the final grading takes place. Any material rejected by the final screen as too large goes to a second roll mill and then returns for screening. That material taken out by the air separator and dust arrester system which contains some coarse particles is put through a tube mill before being bagged. All minus 200 mesh dust is bagged in paper bags for use as filler at the company's roofing plant. The various units are designed and connected as a totally enclosed unit insofar as possible. Arrester pipes are connected into the system at various points such as elevator boots, vibrating screens, etc. It is intended that a slight in-draft be maintained on the equipment as a whole to prevent leakage of dusts into the air.

The first arrester equipment installed was a 21 ft unit used in conjunction with a blower designed to handle 20,780 cfm at a static pressure of 7½ in. of water. The arrester had what is known as a three to one ratio of filter area (*i. e.*, 3 cfm of air handled per sq ft of filter cloth area). This arrester was not entirely satisfactory since the knocker arrangement put too great a strain on the filter cloth, causing many of the bags to break open. The knocker was not capable of keeping the filters clean and the dust accumulated in caked layers until the resistance became so great that the filters stretched. This stretching continued to the point where adjacent filter cloths touched each other on the clean air sides, reduc-

*Dust Control Engineer, Bird & Son, inc.

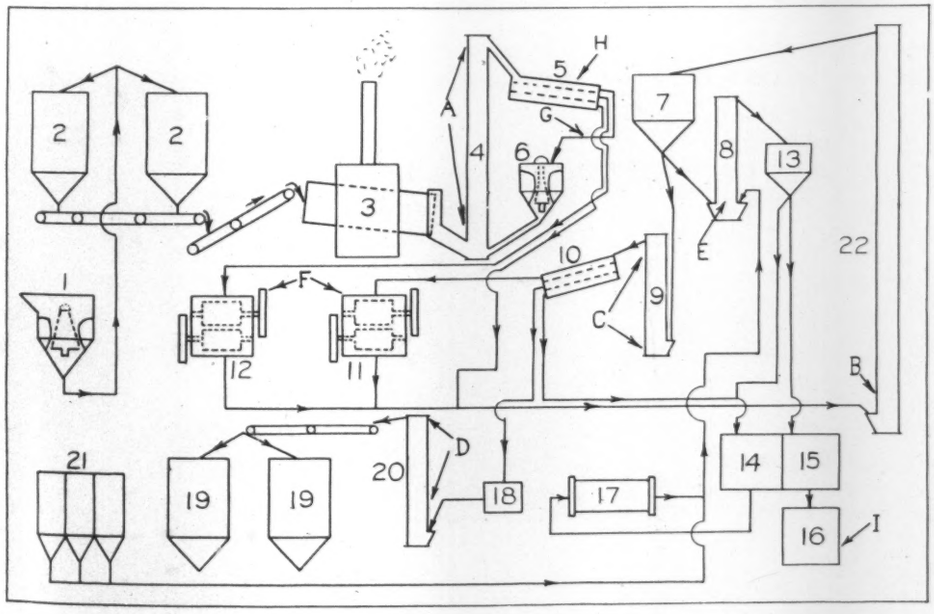


Fig. 1—Flow sheet for Bird & Son rock crushing plant. 1—crusher. 2—rock storage tanks. 3—rock dryer. 4—elevator. 5—scalper screens. 6—crusher. 7—eight foot air separator. 8— and 9—elevators. 10 — vibrating screen. 11 and 12 — roll mills. 13—six foot air separator. 14—junk tank (65 to 100 mesh material). 15—dust tank. 16 — bagging machine. 17—tube mill. 18 — granule scales. 19 — granule storage tanks. 20—elevator. 21—dust arrester. 22 — elevator. Letters on drawing refer to Table 2 and show location of arrester pipes referred to in table

ing the filter area. Additions to the mill necessitated the installation of the present equipment. This consists of a 36 ft arrester unit with a 10 ft classifier section. The old exhaust fan was replaced with a blower capable of handling 25,650 cfm at a static pressure of 7½ in. of water, driven by a 50 hp, 550 volt motor with a full load speed of 1170 rpm and a fan speed of 1520 rpm. The arrester has 12,700 sq ft of filter cloth area, and a ratio of two to one.

Procedure in Taking Dust Counts

When this installation was completed dust counts taken with the Smith-Greenburg impinger ranged from 0.5 million to 6 million particles per cu ft. Since then we have made periodic dust counts with the impinger and also a microscope instrument which have been consistently below 10 million and often as low as 2 to 5 million per cu ft. In order to keep the counts in this low range it is necessary to make frequent inspections and tests on the arrester system. This is done weekly and a report is given to the maintenance department so that necessary repairs may be made during the week-end. The following is our procedure:

1. Dust counts are taken throughout the mill at several places near where the men regularly work.
2. Manometer readings are taken at various places in the piping system and especially at the arrester house where the resistance across the filters is noted.
3. A careful inspection is made of all arrester piping and chutes for any leaks where dust might get into the air. In dark or shadowy places a multi-cell focussing flashlight is a great help in locating leaks.
4. About once a year an outside engineer is called in to make dust counts and a survey of conditions. This is to check results and for a matter of record.

Measuring Static Pressure

Measurement of the static pressure with a water manometer has no particular value in determining the air flow at any point unless the velocity pressure is also known. This may be obtained by using a Pitot static tube, but if a little dust gets into the tube the readings are worthless. A better method is to use a thermometer anemometer which can be made up quite cheaply. The instrument we use, which was designed by C. P. Yaglou of the Harvard School of Public Health, is simply a 220 F thermometer with a wire wound around the bulb and connected to a source of constant voltage (2 to 8 volts). The difference in the temperature reading with the current turned on and that reading when the current is turned off is noted and by referring to a graph which has been previously calibrated the air velocity is read directly.

By manipulating the blast gates in the air ducts the desired velocity can be obtained and the blast gate then locked in position. A static pressure reading can then be taken and used as a standard against which subsequent readings may be checked provided the position of the blast gate is unchanged. In some locations the wear of the blast gates may be excessive, causing the readings to change. The only solution is replacement of the worn part and a recheck with the thermometer anemometer. In using this instrument extreme care must be taken to read the thermometer to the closest half-degree or even to smaller limits if possible. If this is not done, sizeable errors are likely to result.

Air Velocities at Plant

Recent readings of air velocities and corresponding static pressures at nine representative points of ventilation have been made. Total air flow for these nine points is slightly less than half of the total handled by the system. These velocities have also been converted to volumes of air handled by the system and are listed in Table 2.

Section	Velocity, Fpm.	Pipe Size, In.	Remarks
Elevator No. 4	4000	8	1/3 volume to elevator
	4200	5	
Elevator No. 22	4200	8	
Elevator No. 9	4200	5	Velocity taken in 8 in. pipe
	3000	5	
Elevator No. 20	3000	4	
	2500	4	
Elevator No. 8	3000	5	Velocity taken in 8 in. pipe
Roll Mill No. 12	3300	5	
Hood (see photo)	3630	4	
Scalper screen (two)	3540	6	
Bagging mach. hood	2700	12	100 fpm at face 250 to 300 at point origin

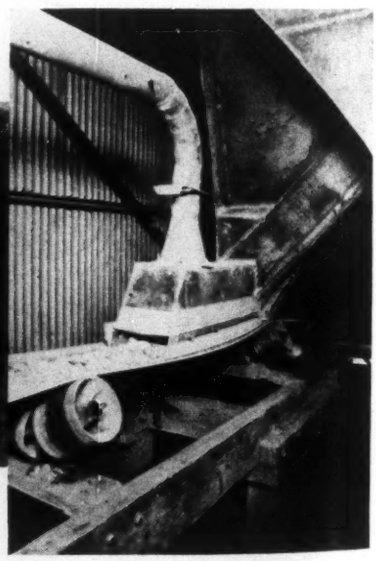


Table 1—Dust Collector Piping, Bird & Son Plant

14 in. pipe	1.05 sq ft
6 in. pipe to crusher	0.196 sq ft
4 in.—Material feeder	0.087 sq ft
4 in.—Material feeder	0.087 sq ft
5 in.—Conveyor hood	0.136 sq ft
4 in.—Top of storage tanks	0.087 sq ft
4 in.—Top of storage tanks	0.087 sq ft
6 in.—Inclined elevator	0.196 sq ft
Total	0.876 sq ft
24 in. main pipe	2.84 sq ft
6 in.—Scalper screen	0.196 sq ft
6 in.—Scalper screen	0.196 sq ft
5 in.—Elevator	0.136 sq ft
5 in.—Elevator	0.136 sq ft
4 in.—Elevator	0.087 sq ft
8 in.—Hummer screen	0.348 sq ft
5 in.—Dust tank	0.136 sq ft
5 in.—Junk tank	0.136 sq ft
5 in.—Junk tank	0.136 sq ft
5 in.—Elevator	0.136 sq ft
5 in.—Elevator	0.136 sq ft
4 in.—Scales hood	0.087 sq ft
4 in.—Conveyor hood	0.087 sq ft
6 in.—Elevator	0.196 sq ft
5 in.—Newhouse crusher	0.136 sq ft
5 in.—Newhouse chute hood	0.136 sq ft
4 in.—Dryer	0.087 sq ft
8 in.—Elevator	0.348 sq ft
5 in.—Crushing rolls	0.136 sq ft
5 in.—Crushing rolls	0.136 sq ft
5 in.—Storage tank	0.136 sq ft
5 in.—Storage tank	0.136 sq ft
Total	3.400 sq ft
14 in. pipe	1.05 sq ft
12 in.—Bagging hood	0.788 sq ft
5 in.—Conveyor	0.136 sq ft

on the velocity showed it to be 2200 fpm. The piping layout has since been changed to overcome this clogging by connecting this section to the 14 in. main line pipe, thus shortening materially the distance to the arrester (for the general piping layout, see Table 1). In the pipe leading from the bagging machine hood where the velocity is listed as 2700 fpm there is no settling whatever, which is probably due to the fact that the dust is all minus 200 mesh at this point. Settlings at other sections are found to be of the coarser materials (6 to 100 mesh). The air velocities used are somewhat lower than those commonly reported as necessary for rock dusts. Dust counts, however, indicate that the concentrations in the air are being kept well below recognized standards.

Solid particles require certain minimum velocities for pneumatic conveying which vary with the size, shape, and density of the particle. For vertical conveying of average solid particles the following equation of Dalla-Valle[1] gives a minimum value for the velocity V in feet per minute:

$$V = 54700 \left(\frac{S}{S+1}\right) P^{0.37}$$

where S = Specific gravity of particle = 3.0 for rock crushed at Bird & Son.

P = particle diameter in feet.

Since we are making 10 to 28 mesh material we do not want to collect through the arrester system any particles coarser than those retained on 35 mesh (0.0232 in.) From DallaValle's equation we find that a velocity of 1150 fpm will convey vertically particles as large as 35 mesh but no larger. As already mentioned, the velocities throughout the mill vary from 2200 to 4200 fpm. The present pipe lay-out is such that these velocities must be maintained to insure in-drafts on elevators and other equipment. Therefore, we have to expect that some of the larger size material (6 to 28 mesh) will be carried into the arrester system. This has been found to be the case and only by judicious placing of the arrester pipe connections is it possible to keep this to a minimum.

[Concluded on p. 313]

Table 2—Air Quantities and Miscellaneous Data

Equipment	Size	Volume Cu Ft	Air Exhausted Cfm	Static Pressure In. H_2O	Material Handled	Reference Letter Fig. 1	Pipe Diam. In.
Elevator No. 4	33′ height	86	1035	2.5	2 in. to 100 mesh	A	5 8
Elevator No. 22	58′ height	192	1450	2.25	3/16 in. to 200 mesh	B	8
Elevator No. 9	28′ height	70.5	1085	2.0	3/16 in. to 200 mesh	C	5 5
Elevator No. 20	20′ height	53	475	2.0	10 to 28 mesh	D	4 4
Elevator No. 8	20′ height	53	510	1.75	100 to 200 mesh	E	5
Roll Mill No. 12	36″ x 16″	.....	450	2.25	1/4 in. to 10 mesh	F	5
Hood over belt conveyor	See photo	1.6	315	2.0	2 in. to 1/4 inch	G	4
Scalper screens (2)	3′ x 8′	45	695	2.0	2 in. to 100 mesh	H	6 6
Bagging machine hood No. 16	20.7 sq ft area	.....	2120	3.25	200 mesh	I	12

If a close check is made of these velocities with the air quantities in Table 2 several apparent discrepancies appear. This is because the 5 in. pipe to the bottom of elevator No. 9 and the 5 in. pipe to elevator No. 8 are only small sections of 5 in. elbow which join in an 8 in. line; therefore the velocity reading was taken in the 8 in. pipe. At another point the 8 in. pipe to elevator No. 4 is connected at the junction of two chutes with the bottom of the elevator. It was assumed, therefore, that the air drawn from the elevator was one-third of the air handled through the 8 in. pipe.

Some settling occurs in the piping but what little there is is taken care of by frequent checkups on the condition of the pipes. One small section of piping at the extreme end of the 24 in. line was clogging continually and a check

Left to Right

Fig. 2—Bagging machine hood with side opened back to show point of origin of dust at outlet . . . Fig. 3 — Large primary crusher. Note absence of dust. Arrester piping is connected to the under portion and draws much of the dust down through the opening . . . Fig. 4—Hood over belt conveyor (*G* on Fig. 1)

[1] "Fundamentals Relating to the Design and Operation of Exhaust Systems," American Standards Association. Also published in *A.S.H.V.E. Journal Section*, Heating, Piping and Air Conditioning, September, 1932, pp. 639-641.

Practical Suggestions for Reducing Piping Corrosion

IN many piping systems the mechanical installation itself has an important bearing on the service life obtained from the pipe. The mechanics of the system as well as the selection of the kind of pipe to be used can be accurately controlled, but in most cases the operating conditions are subject to little or no control. Obviously it is desirable to take every precaution possible that will increase the service life of a given installation and it is with that in mind that the following practical suggestions are offered.

1. Where threaded joints are used the pipe threads should be cut accurately. Pipe with wavy, shaved, rough or non-concentric threads should never be installed as it can and probably will be a source of trouble. Corrosion failures at joints, as well as joint leaks, can be caused by improperly cut threads.

2. When making up threaded joints the pipe threads should be drawn up tight into the coupling or other fitting. In corrosive services the effects of corrosion may be noticed first at the joints, particularly if the threads are not fully engaged or if the joint is not tight due to improperly cut threads.

Properly made joints increase both the durability and the strength of the piping installation.

3. The ends of all threaded pipe should be thoroughly reamed to eliminate burrs before the joint is made up. Burrs reduce the capacity of the pipe, cause turbulence and, where suspended solids are conveyed, may cause obstructions to form. Turbulence, particularly, is undesirable because it tends to increase corrosion.

4. In hot water systems it is desirable from the corrosion standpoint to keep the temperature of the water at not more than 140 F. Usually this is possible in domestic hot water systems.

5. Where water is to be heated for domestic or other uses, it is desirable to employ an open or deaerating type heater which will permit dissolved gases in the water to escape.

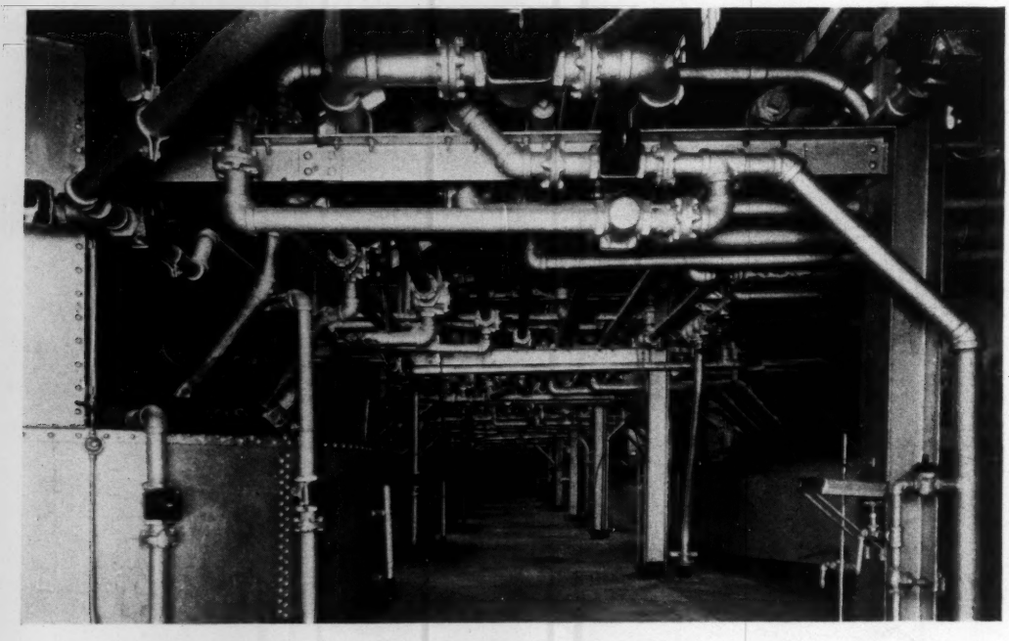
6. For wrought iron pipe and steel pipe it is desirable to use cast iron, malleable iron, cast steel or forged steel fittings, in preference to non-ferrous fittings.

7. Contact between dissimilar metals should be avoided wherever possible and particularly in lines handling solutions which are good electrolytes. The same applies in the handling of hot solutions. However, where ferrous pipe must be connected into non-ferrous fittings the pipe should be extra heavy, or, at least, extra heavy nipples should be placed on each side of and in direct contact with the fittings.

8. In installations where the pipe must be insulated, every effort should be made to provide a continuous covering which will prevent air from reaching the pipe surface. The reason for this is that where air gets beneath insulation, condensation can occur on the metal surface where it becomes acid due to absorption of atmospheric gases and thereby causes rapid corrosion. Furthermore, insulating materials containing acid producing compounds should be avoided.

9. The use of excessive amounts of bowl cleaners, particularly of the acid forming type, should be avoided in order to reduce the effect of corrosion in drainage systems.

10. When installing underground steam lines or other lines subject to wide temperature variations and where such lines are placed in protective conduit, provision should be made for venting the conduit so as to permit free air circulation during the periods when the line is not in use. Otherwise moisture from the air will con-



Wrought iron piping at the Colgate-Palmolive-Peet Co. plant at Jeffersonville, Ind.

From technical bulletin "Wrought Iron for Piping Systems," published by A. M. Byers Co.

dense on the pipe surface where it will dissolve atmospheric gases, thereby becoming acid and causing abnormal corrosion. The vents can be closed when the line is in service.

11. Underground pipe lines that must pass through cinder fills, through soils contaminated with cinders or through soils containing appreciable amounts of sulphur compounds should be installed so that the soil water, which becomes acid through contact with sulphur compounds, can drain away from the pipe and thus minimize the effect of corrosion.

A simple yet effective method of providing for drainage is to place a 4 to 6 in. thick layer of crushed stone (preferably limestone, because it will neutralize the acid) along the bottom of the pipe trench. The pipe should be laid on this base and additional stone should then be placed on each side of the pipe until the pipe is at least half buried in the stone.

As an added precaution, a heavy coat of lime mortar should be brushed on the top of the pipe or, in lieu of that, ordinary lime can be used. The pipe should then be covered and where possible soil free from cinders or other sulphur bearing material should be placed next to the top of the pipe. The balance of the fill can be made using excavated material taken from the pipe trench.

12. Corrosion in the steam return lines of steam heating or power systems can be reduced appreciably by employing several or all of the following suggestions, depending upon the primary source of the trouble.

(a) Reduce the amount of oxygen and carbon dioxide carry-over from the boiler as much as possible through proper treatment and deaeration of the boiler feed water. Most boiler water treatments are intended primarily to keep the boiler clean and free from corrosion and where serious corrosion is encountered in return lines supplementary treatment or deaeration may be necessary.

(b) All joints in return lines operating under subatmospheric pressures should be made air tight.

(c) All horizontal return lines should be given sufficient slope or pitch to permit rapid and complete drainage.

(d) All low points or pockets in the lines where condensate can collect should be eliminated or properly trapped. Occasionally, the pipe supports are so placed as to cause low points in horizontal lines.

(e) Reaming of pipe ends to remove burrs is particularly important in condensate lines.

(f) Cold water injection into return lines ahead of vacuum pumps should be avoided wherever possible.

(g) Extra heavy nipples should be used in return lines.

Dust Control System—

[Concluded from p. 311]

Chip traps are not used at our plant although they sometimes are used to good advantage in preventing large granular pieces from getting into arrester systems. Many times, however, these chip traps have excessive pressure losses which are objectionable.

Alterations and Expansions

The usual method of laying out arrester piping is to add the area of the branch ducts which are to be used and make the main pipe 20 per cent larger than these combined areas. This will give a main pipe velocity 83.4 per cent that of the branch ducts, which should be enough to convey the material.

After a system is in operation it is often necessary to add branch piping because of plant expansion, which results in a change in the ratio of main pipe area to combined branch pipe area. Usually this can be done with good results up to the point where the combined area of the branch piping equals the area of the main line, but beyond this there is a tendency for the velocity in the branch ducts to fall off rapidly. We have had to make additions to our system which have caused the main line area to be less than the combined branch duct areas (see Table 1); as yet we have experienced no trouble because of this. It is quite probable, however, that further additions would overburden the main pipe line to the point where excessive settling would take place.

Filter House Maintenance

About 40 tons of dust are produced daily at our plant, and 95 per cent of this material passes a 200 mesh sieve. The arrester system collects amounts varying from 2 to 10 tons depending upon the wetness of the stone, hardness of the rock, etc., the remainder being handled by the air separators. The large amount handled by the dust arrester at times makes it necessary to run the knocker or bag cleaner frequently in order to keep the resistance across the filters from building up too much. When the cloth filters were new their resistance was 1.5 to 2 in. of water. After they had been in operation for six months a reading showed 4 to 6 in. Since then the resistance has remained about 6 in. before and 3.75 to 4 in. after running the knocker.

With these resistances the fan is operating against a total resistance of from 9 to 11 in. of water. The resulting air volume handled at 1520 rpm should vary from 21,380 to about 16,000 cfm according to the fan manufacturer's data. Actual measurement of the blower exhaust using the thermometer anemometer shows from 23,600 to 19,500 cfm.

We have found it necessary to make frequent inspections inside the filter house so that broken filters and plugged hoppers can be discovered. It is a quite common occurrence for the hoppers under some sections of the filter house, and particularly under the classifier, to plug at the bottom. If this condition is allowed to continue for any length of time the dust will fill the hopper and the air space between those filters directly above the hopper, reducing the filter area tremendously.

Air Conditioning Takes to the Highways

AIR conditioning is taking to the highways to serve the automobile and bus passenger as it has the railroad patron. Elsewhere in this issue is described a completely air conditioned tourist camp, or motor court, in Arizona. And last month, two fleets of air conditioned buses were announced.

Fig. 1 diagrams one of the new fleet of fifty 36 passenger air conditioned buses of Santa Fe Trailways for service between Chicago, St. Louis, Kansas City, Oklahoma and Kansas points, Colorado, and California. The refrigeration equipment—result of six years' laboratory research and road tests—is suspended on rubber mountings beneath the floor. The air conditioning equipment is built into the top of the bus with motor and fans also suspended on rubber mounting. Outside air, drawn through openings on the roof, is filtered, cooled and dehumidified, or heated when the outside temperature is low. Each unit circulates approximately 1100 cfm of air to provide an air change every three minutes.

Efficiency of the system was increased by insulating walls and roof with 1½ in. of insulation and by covering the floor with a ½ in. board of heat and cold proof material. The exterior of the bus is painted with sun glare repellent aluminum paint.

The Interstate Transit Lines (Chicago and North Western Stages and Union Pacific Stages) have equipped 140 buses with year 'round air conditioning equipment for service between Chicago and Kansas City to the Pacific Coast. Fig. 2 illustrates assembly of the refrigerating unit.

Inside temperature is regulated automatically by thermostatic control, and is kept approximately 15 F below outside temperature. Air circulation is one of the unique features of the system. Immediately beneath the steel roof of the bus is a false roof perforated by thousands of tiny holes through which cool air enters

[Concluded on p. 318]

[Fig. 1 courtesy Carrier Corp. Fig. 2 courtesy Baker Ice Machine Co.]

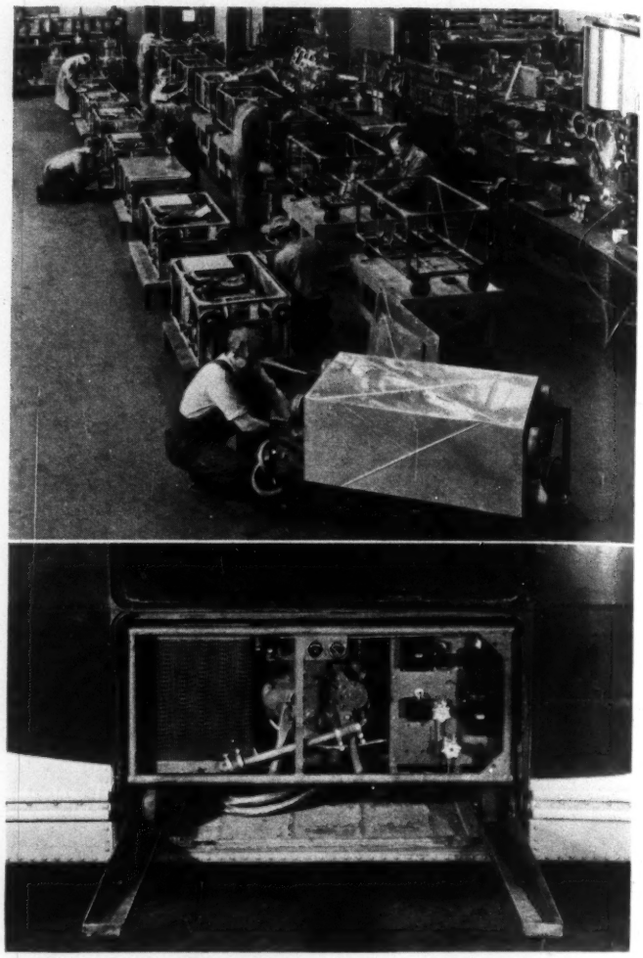


Fig. 2—This unusual picture (*above*) shows the complete assembly process for the unit for the Interstate buses. At the right rear is the empty, tubular frame for the refrigerating power unit. Assembly proceeds step by step across the back row and up the front to show the completed unit at the right front. The lower view shows the unit in regular operating position; as soon as the demountable channel rails are removed and the door is closed, the bus will be all set to go places

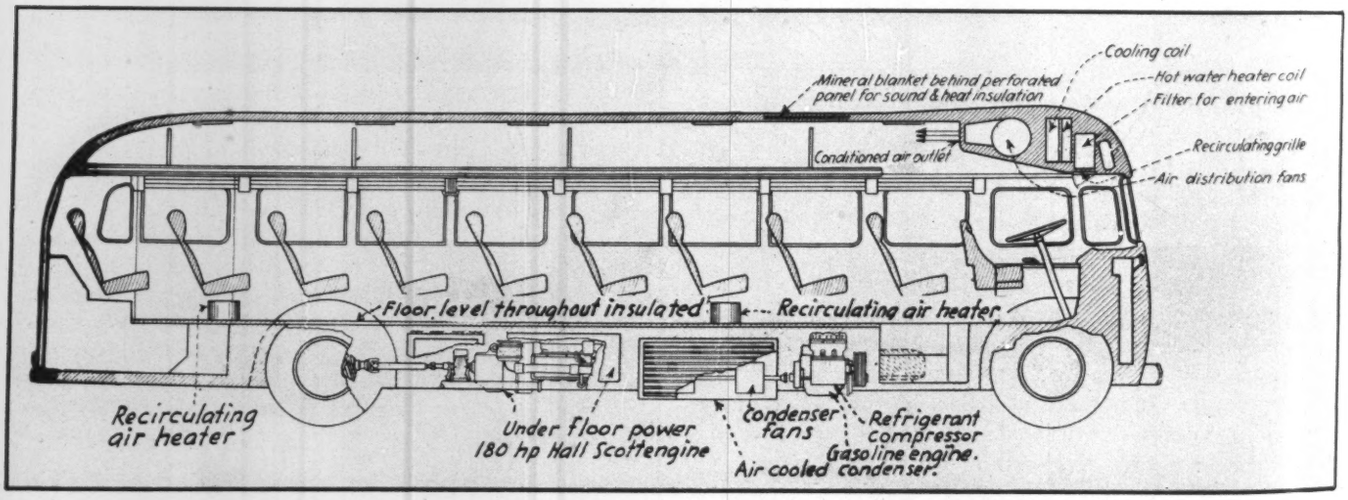


Fig. 1—Diagram of Santa Fe Trailways air conditioned bus

Figuring Solar Heat Gains of Buildings

By William Goodman*

IN many buildings the solar heat gain is a substantial part of the total cooling load. In this article, a series of tables for estimating the probable solar heat gains of buildings is presented. In addition, the theory underlying the computation of these solar heat gains is given, as well as the derivation of the various equations required for the preparation of the tables.

Radiant Energy and Heat

Radiant energy—the energy received from the sun—is not *heat* as we commonly know it. It does not manifest itself as heat until it falls upon an opaque object. A beam of sunlight can be focused through a lens made of ice upon a piece of paper which will burn. Yet the ice lens will not melt. Sunlight may shine through a window upon a dark surface which will become decidedly warm while the glass itself feels relatively cool. The radiant energy of a beam of sunlight is transformed into heat whenever the beam is intercepted by an opaque surface. Not all of the energy of the sunbeam will be converted into heat. If the surface is light in color and smooth in texture, much of the light will be reflected and only a small fraction of the energy of the beam will be retained by the surface as heat. On the other hand, if the surface is dark in color, little of the light will be reflected; practically all of it will be absorbed by the surface and converted into heat.

So far, sunlight has been spoken of as though it consisted only of visible light—that is, light that can be detected by the human eye. Much of the energy in sunlight consists of "invisible light." The eye responds to only a relatively narrow band of wave-lengths. Beyond the visible radiations in a beam of sunlight lie the infra-red rays—invisible but nevertheless responsible for a large part of the heat produced by a beam of sunlight. The infra-red rays act in exactly the same way as the visible portion of the sunbeam. They do not become heat as such until they are intercepted by a solid opaque surface; they are reflected by light colored or bright surfaces and absorbed and transformed into heat by dark surfaces. Therefore, for the purpose of this article, the visible and invisible portions of the solar spectrum may be treated as one.

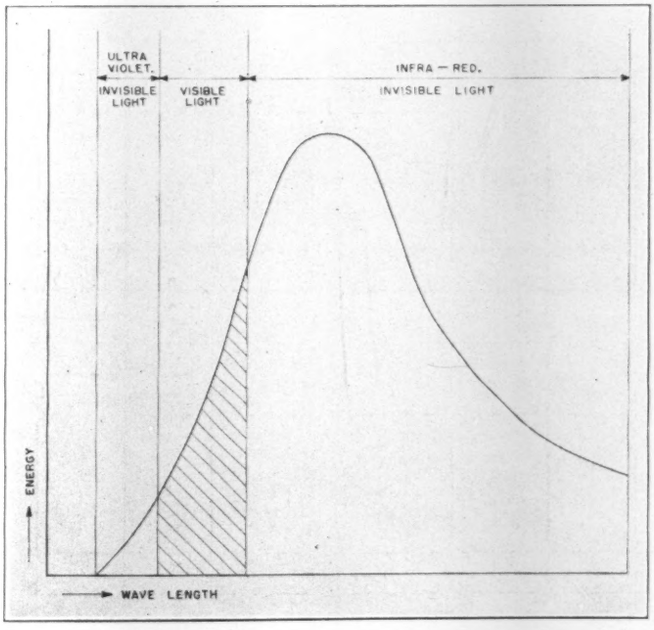
When sunlight passes through a transparent substance such as glass, a small amount of the solar energy is absorbed by the glass and transformed into heat. However, as is shown later, the amount absorbed is only a small fraction of the total energy impinging on the outside surface of the glass; most of this impinging solar radiation is either reflected back to the outdoors by the glass, or transmitted through it into the room. The solar energy transmitted through the glass is not transformed into heat until the beam impinges on the opaque surfaces inside the room—floor, walls, and furniture.

Use of Tables for Windows and Skylights

The solar heat transmitted through bare windows is tabulated in Sections *A* and *B* of Tables 1 to 8.[1] Factors are given later for determining the reduction in solar heat due to shades or awnings. For each latitude, tables are given for both single and double glass.

In using these tables, the time of day at which the heat gain is to be computed and the direction in which the window faces must be known. Frequently the time of day at which solar heat gain is being figured is determined by the character of the installation. For example, in a downtown restaurant doing a large luncheon business, the total heat gain would be ordinarily figured at around noon or one o'clock, thus de-

Solar heat gain represents a substantial part of the total cooling load in many buildings. A series of tables is presented here for estimating the solar heat gains of buildings, and use of the data is explained. . . . . The theory underlying the computation of solar heat gains, and the derivation of the equations upon which the tables are based, will also be given



Curve illustrating approximate distribution of energy according to wave length of radiation. The total energy of each band of wave lengths is proportional to the area under each section of the curve. Judging from these areas, the total energy that can be converted into heat is larger in the region of the invisible infra-red than in the region of visible light

[1] Tables 5-8 will appear with Part 2.

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Part 1.

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termining the time at which the solar heat gain should be figured. In other cases where all of the cooling load except the solar heat gain is steady throughout the day, the time of day is selected at which the *sum* of the solar heat gains through all of the various surfaces is a maximum. This can be ascertained only by actually calculating the solar heat gain at various hours of the day. While this sounds rather formidable, a little experience will soon enable the user of these tables to select the time of day in one or, at the most, two trials.

The solar heat gain through all the surfaces of a building should be figured for the same hour. For example, if windows face both south and west, the solar heat gain through all the windows should be figured at the same time of day.

The time tabulated in Tables 1 to 8 is true solar time. Usually solar time does not differ from standard time by more than about 30 minutes. For most purposes, the values in the tables may be regarded as standard time. If desired, standard time can easily be converted into solar time.[2]

The values in the tables are not the actual total heat gains through glass. As is shown later in spite of the fact that sunlight is streaming through a bare window,

[2]W. Bartky, "Highlights of Astronomy," p. 36. University of Chicago Press.

Solar Heat Gains Through Windows and Skylights and Solar Temperature Differences for Walls and Roofs

Table 1—0 Degrees Latitude

WINDOWS AND SKYLIGHTS

Solar Heat Gain
Btu per Sq Ft of Glass

A. Single glass—Bare

Time → / ↓	Direction Window Faces NE	E	SE	N	Flat Skylight	Time
6 a.m.	50	52	20	14		6 p.m.
7 a.m.	143	147	53	42	23	5 p.m.
8 a.m.	191	190	58	61	98	4 p.m.
9 a.m.	184	170	33	68	170	3 p.m.
10 a.m.	151	116	5	72	225	2 p.m.
11 a.m.	98	40		73	257	1 p.m.
12 M	40			74	269	12 M
1 p.m.	3					11 a.m.
	NW	W	SW	N	Flat Skylight	← ↑

B. Double glass—Bare

Time → / ↓	Direction Window Faces NE	E	SE	N	Flat Skylight	Time
6 a.m.	44	46	14	9		6 p.m.
7 a.m.	125	129	38	27	11	5 p.m.
8 a.m.	167	166	37	39	75	4 p.m.
9 a.m.	158	144	16	44	144	3 p.m.
10 a.m.	124	89	1	46	197	2 p.m.
11 a.m.	70	20		47	226	1 p.m.
12 M	20			48	237	12 M
	NW	W	SW	N	Flat Skylight	← ↑

WALLS AND ROOFS

Solar Temperature Differences
Degrees F.

C. Dark Colored

Time → / ↓	Direction Wall Faces NE	E	SE	N	Roof	Time
6 a.m.	5	5				6 p.m.
7 a.m.	27	28	8	6	2	5 p.m.
8 a.m.	39	39	12	12	20	4 p.m.
9 a.m.	38	35	7	14	35	3 p.m.
10 a.m.	32	24		16	48	2 p.m.
11 a.m.	21	9		16	55	1 p.m.
12 M	9			16	58	12 M
	NW	W	SW	N	Roof	← ↑

D. Medium Colored

Time → / ↓	Direction Wall Faces NE	E	SE	N	Roof	Time
6 a.m.	2	2				6 p.m.
7 a.m.	20	20	5	3		5 p.m.
8 a.m.	29	29	7	8	14	4 p.m.
9 a.m.	28	26	4	10	26	3 p.m.
10 a.m.	23	17		11	36	2 p.m.
11 a.m.	15	6		11	41	1 p.m.
12 M	6			11	44	12 M
	NW	W	SW	N	Roof	← ↑

Table 2—10 Degrees Latitude

WINDOWS AND SKYLIGHTS

Solar Heat Gain
Btu per Sq Ft of Glass

A. Single glass—Bare

Time → / ↓	Direction Window Faces NE	E	SE	N	Flat Skylight	Time
6 a.m.	50	52	20	14		6 p.m.
7 a.m.	137	147	59	32	33	5 p.m.
8 a.m.	176	190	76	37	113	4 p.m.
9 a.m.	159	170	58	31	184	3 p.m.
10 a.m.	116	116	25	27	240	2 p.m.
11 a.m.	59	40	1	22	270	1 p.m.
12 M	10			20	282	12 M
	NW	W	SW	N	Flat Skylight	← ↑

B. Double glass—Bare

Time → / ↓	Direction Window Faces NE	E	SE	N	Flat Skylight	Time
6 a.m.	44	46	15	9		6 p.m.
7 a.m.	120	129	44	19	20	5 p.m.
8 a.m.	153	166	53	20	90	4 p.m.
9 a.m.	133	144	35	14	158	3 p.m.
10 a.m.	89	89	10	11	210	2 p.m.
11 a.m.	35	20		8	238	1 p.m.
12 M	2			7	249	12 M
	NW	W	SW	N	Flat Skylight	← ↑

WALLS AND ROOFS

Solar Temperature Differences
Degrees F

C. Dark Colored

Time → / ↓	Direction Wall Faces NE	E	SE	N	Roof	Time
6 a.m.	5	5				6 p.m.
7 a.m.	26	28	10	4	4	5 p.m.
8 a.m.	36	39	15	7	23	4 p.m.
9 a.m.	33	35	13	7	38	3 p.m.
10 a.m.	25	24	6	6	51	2 p.m.
11 a.m.	13	9		5	58	1 p.m.
12 M	1			5	61	12 M
	NW	W	SW	N	Roof	← ↑

D. Medium Colored

Time → / ↓	Direction Wall Faces NE	E	SE	N	Roof	Time
6 a.m.	2	2				6 p.m.
7 a.m.	19	20	6	2	2	5 p.m.
8 a.m.	26	29	10	4	16	4 p.m.
9 a.m.	24	26	8	4	28	3 p.m.
10 a.m.	17	17	3	3	38	2 p.m.
11 a.m.	9	6		2	44	1 p.m.
12 M				2	46	12 M
	NW	W	SW	N	Roof	← ↑

the temperature of the glass may be so low that there is still some conduction through the glass from the warm outdoor air. For this reason, the conduction heat gain should be added to the solar heat gain in order to find the total heat gain through the glass. The conduction heat gain through a window, wall, or roof in the shade is figured by means of the following familiar equation:

$$H_c = AUD \quad \ldots\ldots [1]$$

where

H_c = heat gain due to conduction through exposed surface, Btu per hr;

A = area of exposed surface, sq ft;

U = overall coefficient of heat transfer, Btu per hr, per sq ft, per degree Fahrenheit difference;

D = difference between temperatures of outdoor and room air.

All of the glass surfaces in a building can be figured for the ordinary conduction heat gain as usual. After this has been done, the solar heat gain can be figured for only those windows through which the sun is shining at the time of day selected. The solar heat gain is found by multiplying the values in the tables by the area of the glass. The time of day in the left-hand column of each of Tables 1 to 8 is used only with the directions in the top row; the time of day in the right-hand column, only with the directions in the bottom row.

Example 1: The bare, single glass windows of a building located in a latitude of 40 deg face in the directions noted below.

Solar Heat Gains Through Windows and Skylights and Solar Temperature Differences for Walls and Roofs

Table 3—20 Degrees Latitude

WINDOWS AND SKYLIGHTS

Solar Heat Gain
Btu per Sq Ft of Glass

A. Single glass—Bare

Time → / ↓	Direction Window Faces NE	E	SE		Flat Skylight	Time
6 a.m.	49	52	21		2	6 p.m.
7 a.m.	130	147	68		43	5 p.m.
8 a.m.	159	190	95		123	4 p.m.
9 a.m.	131	170	87		192	3 p.m.
10 a.m.	79	116	58		246	2 p.m.
11 a.m.	22	40	18		275	1 p.m.
12 M					286	12 M
	NW	W	SW		Flat Skylight	← ↑

B. Double glass—Bare

Time → / ↓	Direction Window Faces NE	E	SE		Flat Skylight	Time
6 a.m.	43	46	16			6 p.m.
7 a.m.	114	129	52		28	5 p.m.
8 a.m.	136	166	72		101	4 p.m.
9 a.m.	106	144	62		166	3 p.m.
10 a.m.	53	89	35		216	2 p.m.
11 a.m.	8	20	6		243	1 p.m.
12 M					253	12 M
	NW	W	SW		Flat Skylight	← ↑

WALLS AND ROOFS

Solar Temperature Differences
Degrees F

C. Dark Colored

Time → / ↓	Direction Wall Faces NE	E	SE		Roof	Time
6 a.m.	5	5				6 p.m.
7 a.m.	24	28	11		6	5 p.m.
8 a.m.	32	39	19		25	4 p.m.
9 a.m.	27	35	18		40	3 p.m.
10 a.m.	17	24	13		52	2 p.m.
11 a.m.	5	9	4		60	1 p.m.
12 M					62	12 M
	NW	W	SW		Roof	← ↑

D. Medium Colored

Time → / ↓	Direction Wall Faces NE	E	SE		Roof	Time
6 a.m.	2	2				6 p.m.
7 a.m.	17	20	7		3	5 p.m.
8 a.m.	23	29	13		17	4 p.m.
9 a.m.	19	26	13		30	3 p.m.
10 a.m.	12	17	8		39	2 p.m.
11 a.m.	2	6	1		45	1 p.m.
12 M					47	12 M
	NW	W	SW		Roof	← ↑

Table 4—30 Degrees Latitude

WINDOWS AND SKYLIGHTS

Solar Heat Gain
Btu per Sq Ft of Glass

A. Single glass—Bare

Time → / ↓	Direction Window Faces NE	E	SE	S	Flat Skylight	Time
6 a.m.	48	52	22		4	6 p.m.
7 a.m.	122	147	77		49	5 p.m.
8 a.m.	140	190	116		129	4 p.m.
9 a.m.	101	170	118	1	194	3 p.m.
10 a.m.	43	116	95	8	244	2 p.m.
11 a.m.	2	40	53	17	272	1 p.m.
12 M			10	20	282	12 M
	NW	W	SW	S	Flat Skylight	← ↑

B. Double glass—Bare

Time → / ↓	Direction Window Faces NE	E	SE	S	Flat Skylight	Time
6 a.m.	42	46	16		1	6 p.m.
7 a.m.	106	129	61		34	5 p.m.
8 a.m.	117	166	93		106	4 p.m.
9 a.m.	75	144	93		168	3 p.m.
10 a.m.	22	89	69	2	214	2 p.m.
11 a.m.		20	30	6	239	1 p.m.
12 M			2	7	249	12 M
	NW	W	SW	S	Flat Skylight	← ↑

WALLS AND ROOFS

Solar Temperature Differences
Degrees F

C. Dark Colored

Time → / ↓	Direction Wall Faces NE	E	SE	S	Roof	Time
6 a.m.	4	5				6 p.m.
7 a.m.	23	28	13		8	5 p.m.
8 a.m.	28	39	23		26	4 p.m.
9 a.m.	21	35	24		40	3 p.m.
10 a.m.	10	24	20		52	2 p.m.
11 a.m.		9	12	3	59	1 p.m.
12 M			1	5	61	12 M
	NW	W	SW	S	Roof	← ↑

D. Medium Colored

Time → / ↓	Direction Wall Faces NE	E	SE	S	Roof	Time
6 a.m.	2	2				6 p.m.
7 a.m.	16	20	8		4	5 p.m.
8 a.m.	20	29	16		18	4 p.m.
9 a.m.	15	26	17		30	3 p.m.
10 a.m.	6	17	14		39	2 p.m.
11 a.m.		6	8	1	44	1 p.m.
12 M				2	46	12 M
	NW	W	SW	S	Roof	← ↑

Find the solar heat gain through the windows.

Windows in east wall: 175 sq ft.
Windows in south wall: 350 sq ft.
Windows in west wall: 250 sq ft.

Solution: Inasmuch as the total area of the west and south windows is greater than the area of the east and south windows, the maximum solar heat gain will occur at some time during the afternoon. Therefore, try 3 p. m. Turning to Table 6A,[1] at 3 p. m. the solar heat gain through the south windows is 18 Btu per sq ft, and through the west windows, 170 Btu. The solar heat gain through the windows is:

South, 350×18 = 6,300 Btu per hr
West, 250×170 = 42,500 Btu per hr

Total solar heat through windows = 48,800 Btu per hr

The amount of solar heat transmitted through a window with shades or awnings can be computed by multiplying the total solar heat gain through a bare window by the following factors which were obtained by rounding off the experimentally obtained figures given in the A.S.H.V.E. "Guide 1938."

Light colored shades, half drawn	70 per cent
Light colored shades, fully drawn	45 per cent
Light colored venetian blinds	60 per cent
Awnings	30 per cent

Dark shades should be figured the same as bare windows. Little of the impinging radiation is either reflected or re-radiated through the glass from the dark surface.

Direct sunlight should be excluded from conditioned rooms. Direct sunlight falling on people will cause discomfort even in a conditioned room.

Example 2: Take the conditions of Example 1 and find the solar heat gain through the windows if venetian blinds are used.

Solution: From Example 1, total solar heat gain through bare windows at 3 p. m. = 48,800 Btu

Total solar heat gain through windows with venetian blinds = $0.6 \times 48{,}800 = 29{,}280$ Btu per hr.

Although double glass is not common in air conditioned buildings, nevertheless it is being used to some extent, particularly to avoid condensation of moisture on window panes during the winter months. The data in the *B* sections of the tables are useful for computing the solar heat gains through double glass windows.

When the rays of the sun are perpendicular to the surface of a sloping skylight, the solar heat gain through such a skylight will be greater than through either horizontal or vertical glass. When the rays of the sun are perpendicular to a glass surface inclined at 45 deg, the solar heat gain will be about 40 per cent greater than through vertical glass facing in the same direction ($1 : \cos 45 = 1.4$).

The use of the tables for computing the solar heat gains through walls and roofs will be discussed next month.

[To be continued]

to the Highways—

[Concluded from p. 314]

the coach, changing the air every three minutes. When the air reaches the floor, it is removed through vents.

The system consists of four principal units—the air conditioning refrigeration power unit, the condenser, the cooling unit and the master control station. The self powered, automatically controlled air conditioning unit is self contained and consists of a framework of welded aeroplane tubing incorporating a light weight, moderate speed, four cylinder compressor complete with self starter, governor, and generator linked by multiple V belts.

This assembly is mounted on rubber tired wheels to eliminate road shock and to permit the unit to be withdrawn on demountable channels for inspection. Flexible connections make possible inspection of the entire unit while in actual operation, and the unit can be removed altogether when cold weather arrives. The condenser is of the evaporative type. A separate water supply tank is used.

The generator supplies power for the condenser fan and pump motor, as well as the fan for engine radiator cooling. A control panel mounts necessary controllers, starting switches, high pressure cut-out, oil and temperature gages, etc.

Cooling units were designed to take the least possible space in the ceiling of the bus, since maximum head room must be maintained. They are mounted in the rear, where outside air is brought from grilles in the front of the bus. Air is forced into the space above the false ceiling by scroll type fans with power from the lighting batteries. Returning air reaches the cooling unit through grilles in the ceiling at the rear of the bus. Two filters are used, one for dust and one for odors.

Control of the system is from the driver's master control station at the front of the bus where both visible and audible signals indicate its functioning. An automatic load control is provided to automatically increase the capacity of the refrigerating unit under increasing heat loads. A selector control insures proper sequence of all functions of the unit during starting and stopping of it.

Preventing Occupational Disease

Two hundred industrial concerns support the research and studies of the Air Hygiene Foundation for the prevention of occupational disease, H. B. Meller, managing director, announced recently. During 1937, six states passed laws compensating workmen for occupational diseases, he pointed out. Twenty-one states and the District of Columbia now have such laws, and eight other states are considering enacting this legislation. In the next few years all of the industrial states will probably have passed laws for this purpose. We must seek ways and means of stringently curbing or controlling industrial diseases, says Mr. Meller. Otherwise industry and the community may be burdened with mounting compensation costs, to say nothing of the loss in earning and buying power of the affected workman and his dependents.

Continuing its efforts to aid prevention of silicosis, the U. S. department of labor is releasing a motion picture depicting the harmful effects of the disease and the methods by which it may be prevented. Entitled "The Story of Silicosis," it is being produced under the direction of R. Campbell Starr, secretary of the National Silicosis Conference and a labor department staff member.

Hot Water System of Advantage in Meeting Plant Heating Requirements

By Samuel R. Lewis*

IN the summer of 1935 we were called on to inspect and report on the heating for a large group of buildings for the American Manganese Steel Co. at Chicago Heights, Ill. The plant includes foundries with sand preparation departments, a large pattern shop with an extensive storage building for patterns, machine shops, shipping building, and many small detached and semi-detached auxiliary buildings such as shop offices, laboratories, etc.

Part of the plant had been heated by direct steam, with return mains buried in the ground. Many of the radiators and pipe coils were poorly graded due to having been changed from time to time because of additions and alterations in the machinery and building structures. The major foundry buildings were unheated except for open salamanders burning coke. The old boiler plant was completely worn out, especially because the available deep well water carries a heavy percentage of precipitable solids.

The author—a leading exponent of the use of water as a heat conveyor in heating or cooling—illustrates the advantages this kind of system offers in many types of industrial plants . . . The system described is an overhead forced circulation high temperature layout using unit heaters under automatic control. The rather unconventional procedure of circulating the water directly through the boilers was adopted, and the boilers have additional tubes in what would normally be the steam space

A first consideration in the design of the new heating plant was the elimination of any heating system pipes buried underground where the inevitable corrosion eventually would destroy them. Another reason for eliminating the underground pipes was that all floors in the foundries are of dirt and are constantly being wetted; also, the heavy castings occasionally are dropped —none too gently—by the crane men, which would damage buried piping.

The main buildings all are lofty and all are served by long-span traveling cranes. There is no space, therefore, for conventional radiators or pipe coils close to the floor except along exterior walls, many of which are merely of corrugated steel, often temporary in nature and offering no protection to pipes or heaters.

In order to reduce the smoke, water vapor and dust always present in a steel foundry, a large amount of ventilation is required, using air from outdoors, with adequate provision for control of the temperature of the entering air. If cold air from outdoors comes in contact with the hot moist air rising from the sand around a freshly poured casting, a dense fog will form. This condition is objectionable because it obscures vision and slows down plant operations.

*Consulting Engineer. Member of Board of Consulting and Contributing Editors.

High Temperature Hot Water Adopted

These considerations of bad water, unfavorable floor and wall space conditions, and need for ventilation led to the adoption of an overhead forced circulation high temperature hot water heating system, using electric unit heaters of centrifugal fan type under automatic control. Many of these heaters have air inlets from outdoors and are equipped with air filters for reducing the dust concentration inevitable in the recirculated air and even in the outdoor air near a heavy duty foundry.

Water Circulated Directly Through Boilers

Since there is no need for process steam, since the water all can be recirculated, and since the water is unfavorable for steam generation in any event, the rather unconventional procedure of circulating the water directly through the boilers was adopted. The total load to heat the general plant to 50 F in —10 F weather (offices and toilets and lockers of course to 70 F) is

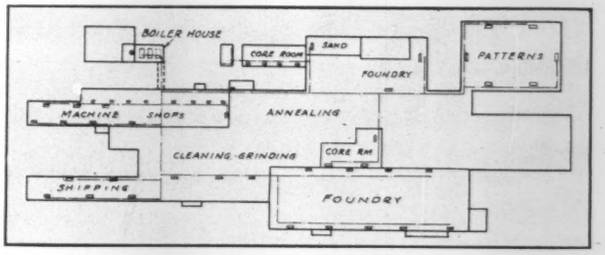


Fig. 1—General plan of heating system

10,000,000 Btu per hour. Two 72×18′ 0″ high pressure horizontal return tubular boilers were installed, and space was allowed for a probable future third similar boiler. Additional tubes were installed in what normally is the steam space above the water line, which increased the heating surface by about 10 per cent.

The boilers are suspended, each with 8 ft between the bottom of the shell and the floor of the combustion chamber. The combustion chambers are designed for burning preheated No. 6 industrial fuel oil, using compressed air atomization, with a heat release of about 11,000 Btu per hr per cu ft of combustion chamber volume. The usual safety devices and automatic con-

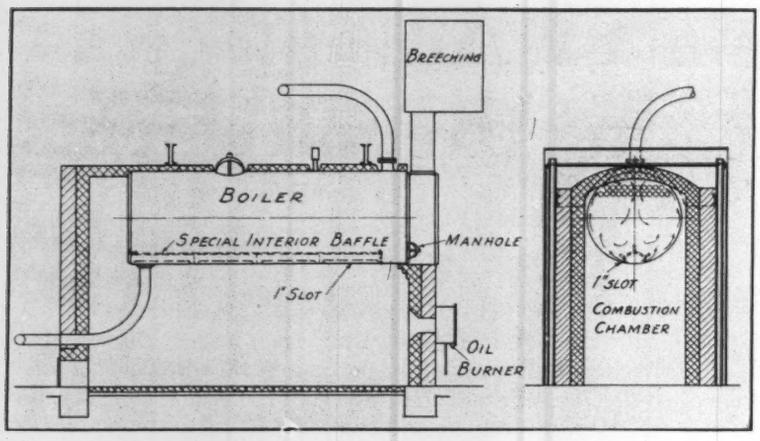


Fig. 2—Sections through one of the boilers

trols for the oil burners are provided, including "electric eyes" to insure stopping of the oil injection in case of ignition failure. The oil burning equipment is in duplicate.

The two underground oil tanks have an interesting steel dome over the manholes and tank heater piping, permitting unusual ease of maintenance. The oil preheater in each tank comprises a seamless spiral pipe coil encircling the oil suction and return pipes, all arranged for easy removal from the tank through a special manhole. Hot water from the general heating system is pumped through these tank heating coils and the oil is warmed very successfully by them.

Boilers Have Special Baffles

Since there was some question about the path of the pumped water from inlets to outlets of such large boilers, and some apprehension of short circuiting due to the very rapid water flow under mechanical circulation, each boiler was provided with a special interior baffle running from the bottom inlet at the rear end, to the front end. This baffle is in sections and easily can be removed from the boiler through the front head manhole. It causes the entering water to flow in a high velocity stream along the bottom of the shell above the hottest part of the fire, and to sweep up the sides of the boiler shell in thin high velocity streams. In operation this controlled and induced water circulation has proved effective and satisfactory. It seems (observation of course is difficult to prove) to induce a downward secondary water circulation through the spaces between the tubes. Anyway, the water becomes as hot as desired, and the system apparently operates with economy and efficiency.

In such a large plant there is of course some continual loss of water through air vents, pump glands and the like, though the need for make-up water is small. In order to reduce as much as possible the fouling of the system with the magnesia-lime concentrates, a closed feed water heater is provided so that by heating the water to its critical temperature for precipitation outside the boilers, much of this debris is deposited before the water enters the circulating system. The tube exteriors of this heater can be cleaned occasionally and the tube interiors receive the generally clean water of the boilers.

Arrangement of Mains

The flow mains drop from above the boilers into a concrete tunnel crossing a wide driveway and then rise so as to pass above all craneways. The return mains parallel the flow mains within the buildings, both almost always being far above the unit heaters, pipe coils, radiators, etc., which they serve. The return mains in the boiler house rise out of the tunnel and reach duplicate circulating pumps of the centrifugal, direct connected type, which discharge the water through the boilers. One pump and one boiler ordinarily serve the entire plant, since there is such a wide range of possible water temperature and since there is a considerable heat storage against sudden outside temperature fluctuations.

The expansion tank is of the pneumatic type, and the entire water carrying system, including the boilers, can operate, if necessary, at 100 lb or more pressure, theoretically permitting a water temperature as high as 330 F. This of course is unnecessary, but a pressure of 50 lb is common on very cold days, giving a water temperature close to 300 F. Since the water mains are higher in most cases than the heat transmitting surfaces, automatic air venting is a simple and inexpensive process. Since the buildings are of variable height, the mains run up or down as convenient, with of course adequate consideration of venting and drainage problems. Wherever the mains cross below monitor windows where the insulation might become wet from rain or snow, the insulation has a waterproof jacket.

Soon after the new plant went into service an unaccountable loss of water was observed. It developed that some of the workmen, discovering that hot water was available by opening one of the drain valves for a low point in the system, were surreptitiously washing their automobiles! It thus proved necessary to padlock all accessible drainage valves.

Fig. 3—Typical cross section through a foundry building

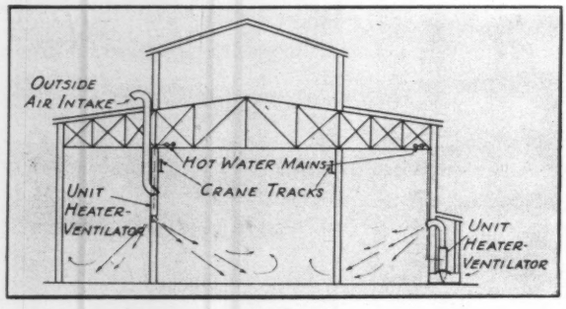
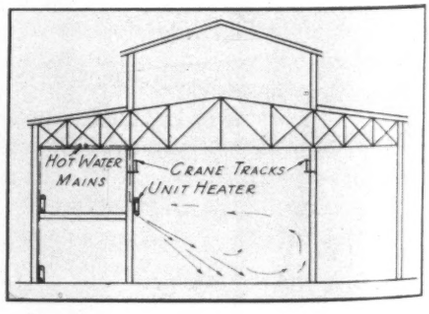


Fig. 4—Typical cross section through a machine shop



Flexibility of System an Advantage

The original design water temperatures being conservative, it is possible with a hot water heating system of this sort to "overload" the pipes to a considerable extent simply by increasing the water temperature, permitting a wider spread between the outgoing and returning currents. The remarkable flexibility of hot water circulation appears to be particularly fortunate in large industrial applications where alterations and additions constantly are being made. Exact maintenance of pipe and heater level or grade is of no particular consequence with a forced water system, and maintenance of traps, reducing valves and the like is eliminated.

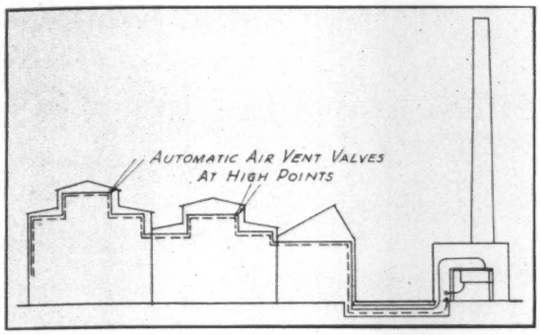


Fig. 5—This gives some idea of the vertical offsets in hot water flow and return mains practicable in a large plant

The absence in large main crane bays of return pipes along columns and under floors is a desirable feature. Hot water unit heaters are no more trouble to maintain than steam unit heaters. Maintenance of unit heater fan bearings, electric motors and air filters in any large foundry is a serious item due to the abrasive dust, but on the other hand there are offsetting objections with any other type of heating system in such buildings.

It has been found, during non-freezing intermediate seasons in the spring and fall, that the heat storing characteristic of hot water is favorable to long periods of non-operation, as overnight. A water system cools very slowly when the pumps and fans cease to operate, and the contraction and expansion strains on the piping system are gradual and rather gentle. These characteristics are very favorable in the heating systems for large industrial plants.

The outstanding reflection of many years' experience in heating industrial buildings is the value to the owner of types of roof and wall construction which will resist heat transfer. In one recent survey it was clear that the owner, who burned a costly fuel efficiently but who thought he needed an increased heating system for a very large and tall factory room, was entirely mistaken, since the same investment applied to insulate a concrete roof made a larger heating system unnecessary.

Check Charts for Air Conditioning Calculations

By J. M. Dean*

IN connection with my duties as an air conditioning engineer for an electric power company, I have endeavored to reduce the work of checking air conditioning jobs to a simple routine procedure. The forms and charts to be presented here were prepared to provide a means of quickly checking computations to approximate slide rule accuracy, and to establish a procedure to serve as a guide for such calculations.

Unless one is working with these computations quite regularly, the form of the work is likely to be forgotten, with the result that the work may become rather tedious. Furthermore, a simplified graphical form lends encouragement to check smaller jobs and prospective jobs which time might not permit to make complete checks upon if one had to go through all the computations.

The computations covered are those involving the survey, the breakdown of the heat gain, the volumes and psychrometric conditions of the various quantities of air to be handled, including the saturation temperature of the air leaving the coils, amount of air to be passed through the coils, amount to be bypassed, and sensible and latent capacity checks. The computations bring one up to the point of actual selection of equipment.

For the most part the charts are in nomographic or "alignment chart" form which, with the use of a straight edge and a hard, sharp pencil, permits the immediate solution of equations containing three variables when any two of them are known. Some use is also made of straight linear conversion scales, for example watts to Btu per hr, cfm to cfh, etc. The arrangement of the charts is in logical sequence so that each value is computed as required for the next step.

The range of the scales is such that jobs up to 50 tons or more can be handled with single computations.

The calibrations of the conversion scales and nomograms are arranged to enable reading quite closely to three significant figures, or slide rule accuracy, and it was considered to be in the interest of consistency to maintain the same relative accuracy in each computation. No originality in material is claimed. These charts simply represent standard data put into graphical form.

The Comfort Cooling Survey Analysis (presented on the following page) is designed to show a complete breakdown of the heat gain along with a complete consideration of all design conditions. Two gain columns are provided, *A* and *B*. As noted, the *A* column is for analysis of the maximum outside high dry bulb condition. The *B* column is for analysis of the maximum outside humidity, mild dry bulb condition. It is well to check the requirements for both conditions before making selection of equipment, and then to pick equipment that will meet both sets of conditions.

The "guide sheet" serves as an index to the charts and as a form for filling in values computed from the charts.

The other charts and tables in this series will follow in order in future issues.

*Air Conditioning Engineer, Des Moines Electric Light Co. All rights reserved.

Check Charts for Air Conditioning Calculations

[See preceding page]

Prepared by _____ Date _____ 19___

_____

COMFORT COOLING SURVEY ANALYSIS

_____ For: _____, Address _____

Conditioned space to be used for _____

Principal dimensions: L___; W___; H___.
Actual Vol.____cu. ft.
No. Occupants_____
Occ. Activity_____
Cond. Water:
Source_____
Max. temp.____deg.
Ventilation Design:
____CFH=____CFM.
Psychrometric Design:
A= max. high DB cond.
B= mild, max. humidity.

	Outside		Inside	
	A	B	A	B
DB				
WB				
TH/#				
RH%				
DP				
G/#				
G/cf				
Cf/#				

Sensible Heat Gains:

S.H. gain thru bldg.

Surface	sq.ft	U	Temp. dif. A	Temp. dif. B	A-load Btu/hr.	B-load Btu/hr.	
No. wall							
" doors							
" glass							
Ea. wall							
" doors							
" glass							
So. wall							
" doors							
" glass							
W. wall							
" doors							
" glass							
Ceiling							
" glass							
Floor							
Misc.							
" "							
Total Bldg. Transmission gains							

Sun gain: Walls_____; Glass_____; Roof_____; Total
Occupants: _____ x _____
Lighting: _____ watts x 3.415
Motors (in conditioned space): _____h.p. x _____
Other internal S.H. components: _____
Total INTERNAL S.H. components
Air Required: _____Btu/hr. Int. S.H. = ___CFM
1.085 x _____ D.T.D.
Fan Motor gain: _____CFM x 0.7
Duct gain: _____Btu/hr. Int. S.H. x 0.04
Ventilation S.H. gain: _____CFH x 0.018 x ___deg. T.D.
TOTAL SENSIBLE HEAT GAINS
Latent Heat Gains: Occupants: ____x_____
Other internal L.H. components: _____
Total INTERNAL L.H. components
Ventilation L.H. gain:
_____CFH x (____G/#out. - ____G/#in.)x 0.011
TOTAL LATENT HEAT GAINS
TOTAL GAINS (all S.H. & L.H. gains)
Tons of Refrigeration
Percentage Total S.H. : Total Gains
Percentage Total L.H. : Total Gains

GUIDE for use of charts to facilitate computations in making analysis of air conditioning survey and in making computations to determine psychrometric conditions in the system, weights and percentages of air, conditions in mixtures, etc.

Chart No.

I. Computations for the analysis:

Area and volume	1
Heat transfer coefficient, "U"	2
Heat transmission through building surfaces	3
Sun gain	4
Occupants: sensible and latent heat components	5
Lighting wattage gain	5
Motors and other heat sources in conditioned space	
Air, in CFM, required for circulation in cond. space	6
Fan motor gain	5
Duct gain	5
CFM-CFH conversion	5
Ventilation gains: sensible and latent heat components	7

II. Computations following the analysis, incident to the design of the system. Enter results in "Design Guide".

Characteristics of air entering conditioned space	8
Total pounds of air to be supplied	9
Characteristics of air entering the fan	10
Pounds per min. air, R.A. and O.A. and percentages	11
Characteristics of air mixtures (air entering coils)	12
Saturation temperature of air leaving coils	13
Percentages of air (by wt.) through coils and by-pass	14
Pounds per min. and CFM air through coils and by-pass	15
Capacity checks	16

DESIGN GUIDE

	DB	WB	DP	%RH	TH/#	G/#	G/CF	CF/#
1. Outside design								
2. Inside design								
3. Air entering cond. space								
4. Air entering the fan								
5. Air entering the coils								
6. Air leaving the coils								

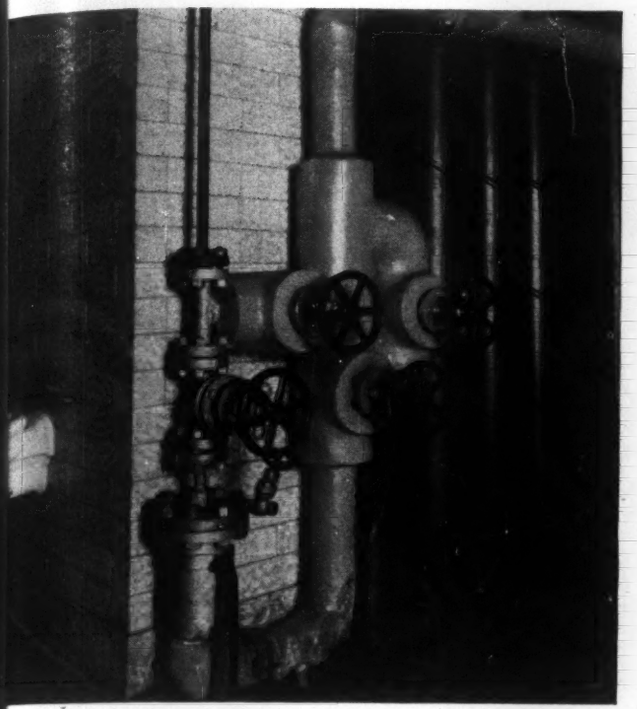
7. TH/# extracted (5. - 6.) _____

8. Total pounds per minute air to be supplied _____#/min.

9. Pounds per min.: Outside air = _____ = _____ %
Recirc. air = _____ = _____ %
Total air = _____ = 100 %

10. Pounds of air through coils and by-pass and CFM through each:

	% (by wt.)	Lb./min.	CFM
Coil air			
By-pass air			



Steam trap on 650 lb line illustrating good layout. Note bleed off line to atmosphere above valves and test cock on discharge of trap

Drips and Drains for High Pressure Piping

By Walter Swanson* and J. W. Hieronymus†

Here's a practical discussion of methods of draining and dripping high pressure, high temperature steam piping. The problem of trapping and dripping high pressure lines is somewhat different from that encountered in connection with low pressure systems. . . . The authors are familiar with several recent installations, base their conclusions on the practices followed on them

DRAINAGE of condensate from high pressure, high temperature steam piping offers a somewhat different problem from that encountered with low pressure steam—particularly low pressure steam having little or no superheat. In addition to the physical properties required of drainage equipment for use with high pressure steam, application of the equipment is not subject to the wide range in selection and location of similar equipment used on low pressure systems. Use of a steam trap in high pressure work in some cases is not only a waste of money, but would be a source of considerable maintenance trouble and expense.

On the other hand, due to high velocity, traps are most important to protect pressure reducing valves, throttle valves, and similar equipment as well as the actual machinery to which the steam piping is connected. Accordingly, the drainage of condensate of high pressure steam requires more serious consideration than is ordinarily given to low pressure steam systems. A thorough and systematic drainage system well repays the study involved.

By high pressure we refer, for the purposes of this discussion, to steam having a gage pressure of 450 lb or over and a total temperature of 750 F or higher. The practice followed in this particular range should also lap over into the field of lower pressures where steam having considerable superheat is used. In fact, it would be wise from the standpoint of safety and of maintenance costs to apply high pressure drainage practice to steam having a total temperature of 450 F regardless of the pressure.

First let us consider the construction or physical properties of high pressure drainage equipment, and then its application to piping systems. The first item of this equipment to be considered is the steam trap itself, and then the valves used in the connection thereof.

*Sargent and Lundy, Inc., Consulting Engineers.
†Barrett-Christie Co.

Construction of High Pressure Drainage Equipment

For high pressure, high temperature work the trap body should be of forged steel of the proper A. S. M. E. specifications for the pressures and temperatures involved. All internal parts except the valve and seats should be of at least 18-8 stainless steel, and no "white metal" should be used in the trap. The valve and seat should be of chrome steel alloy heat treated to a hardness of not less than 52-C Rockwell. However, care should be exercised that the relative hardness of the valve and seat should be such that they will not seize. The recommended difference of hardness between the valve and seat is not less than five on the Rockwell "C" scale.

This, of course, is general valve practice for high pressure, high temperature work and has been found necessary to overcome seizing of valve and seat when these two are of identical metals and identical hardness. Seizing and galling refer to the condition of the valve and seat after the valve has been closed tight, and, due to the high temperature and the pressure of the valve on the seat, these two parts become practically "welded" together—not entirely, but sufficiently so that when the valve is reopened, a portion or sliver of the one will adhere to the other.

The gaskets used should be of soft iron, we believe, having a hardness of not more than 90 Brinell and as uniform as possible with no hard spots which may tend to allow the joint to be pulled up tighter where the softer areas of the gasket are causing an uneven joint. Soft iron gaskets are advised because this material has about the same expansion characteristics as the pipe metal, and is not affected by the heat as much as some other materials.

The bolts and studs used to assemble high pressure traps should possess the tensile strength and elastic limit suitable for the total pressures exerted on the area of the flange or cap they are holding. The bolts or studs should have a continuous thread, as the conventional method of threading in which there are only the threads necessary for the nut with the remaining portion of the stud the full diameter of the stock may cause loosening due to unequal expansion.

When the trap size, (that is, the nominal size of the inlet and outlet pipe connections) increases to the point where the general specification for piping calls for flanged fittings, it is recommended that the trap be so designed as to have the flanges an integral part of the trap body with the female flange on the inlet and the male flange on the discharge side.

The trap should be of simple and substantial construction with as few moving parts as possible, and these parts should be readily accessible.

Trap Capacity

The matter of trap capacity brings us more into the application of the trap than its design. However, if the amount of condensate to be handled by the trap will be consistently a definite and large amount, a straight-through orifice will be satisfactory. But, if the quantity is likely to be small and at irregular intervals, it seems advisable to supply the trap with an orifice (valve and seat) which has the characteristics of a pop safety valve in order to insure positive, intermittent discharge instead of the semi-continuous (dribble) discharge of the straight-through orifice under extremely light load. The semi-continuous discharge of a straight-through orifice under very light loads subjects the valve and seat of the trap to wire drawing, resulting in short life and frequent replacement of parts. By using a restricted or double seated orifice, which in one type is so constructed that the hemispherical valve contacts the larger outer seat but does not quite touch the inner seat, a back pressure will be created, because the condensate cannot pass between the restricted orifice and valve as fast as it enters through the outer seat. This back pressure counteracts the pressure holding the valve to its seat, reducing the effective pressure and allowing the weight of the bucket to pull the valve wide open.

Valves for Trap Installations

The valves used in connection with the installation of the trap, and those used as drains, should be built of the same material as the trap, the body being of forged steel, with stainless steel trim except the valve and seat which should be an alloy heat treated to a hardness of about 52-C Rockwell. The preceding remarks on seizing apply as well in this case. Some difficulty has been encountered where the valve seat is screwed into the body; a slight loosening of this joint has allowed steam to leak through between the valve seat and body, which soon cuts a considerable path for itself, causes a heavy steam loss, and ruins the valve body. To overcome this, for high pressure work a valve having a seat welded into the body so that this seat becomes practically an integral part of the valve body should be used, we believe.

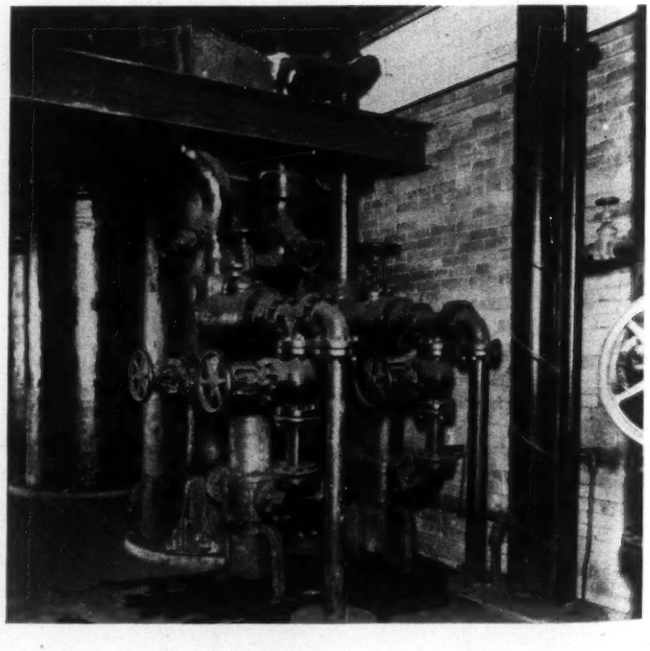
Condensation in Piping

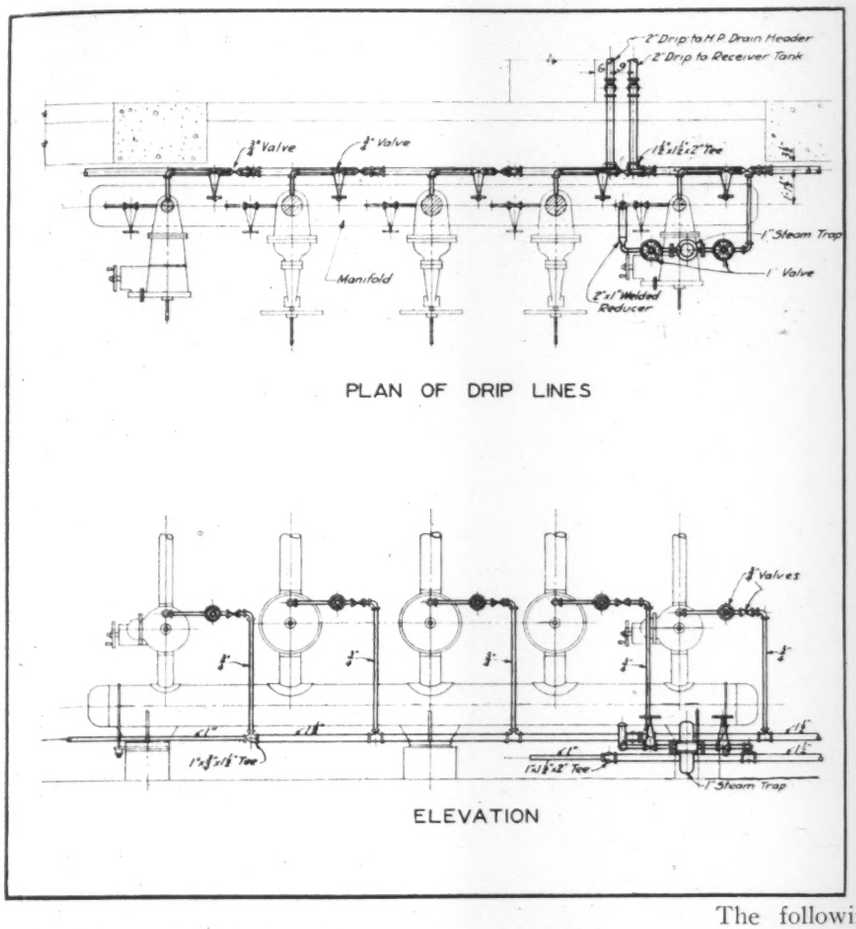
The use of high temperatures practically eliminates the need of steam traps for the purpose of taking care of condensate due to radiation from the piping and equipment, for what little condensate may occur due to radiation from exposed parts of shut-off valves, etc., will be re-evaporated when entering the steam main. Although this reason for traps is less important in high pressure than in low pressure work, the need of a trap in a high pressure line for the removal of condensate created during the heating up period is even greater than in a low pressure system. Condensation will occur in superheated steam lines when the piping system is being warmed up, or when velocities are materially reduced. Because of the weight of the pipe used in high pressure installations and the high temperature to which it must be heated, the trap must have a capacity which will easily take care of this load and protect the equipment in the line.

Unfortunately, comparatively little is known about the flow of condensate in high velocity steam mains. One possibility is that when the steam first condenses, due to contact with the relatively cooler walls of the pipe, it forms a film of water all around the interior surface of the pipe. As the film thickness increases, most of the water flows to the bottom of the pipe to form a tiny stream of condensate.

In spite of the high velocity of the steam, the small trickle of condensate at the bottom of the pipe moves rather slowly because of the friction between the pipe and the water. However, as the depth of the stream increases, the relative friction decreases, and the trickle of condensate accelerates in speed. With increased depth, too, waves will form on the surface of the condensate stream and still further increase its speed.

Heavy duty compound traps at Waukegan Station





Details of drips and drains for a high pressure header

Should these waves attain a size such as to partially or wholly block the passage of steam, trouble will result. Therefore, it becomes evident that it is necessary to drip the steam lines and remove this condensate before it becomes a serious hazard.

Where the piping layout governs the location of the drips because of change of elevation of portions of the piping system, or because of valve locations, or because equipment is to be protected, the problem then is only to figure the amount of condensate to be handled. But where the pipe runs may be several hundred feet in length, with no definite reason for dripping at any particular points, it is usually advisable to install traps at intervals of approximately 400 to 500 ft. This spacing will depend considerably, of course, upon the actual piping layout.

Protecting Valves

Shut-off valves in a piping system should be placed, when practicable, at the highest point in the leads between the boiler and the main header. A drip should, of course, be provided just ahead of a turbine throttle valve. It should be remembered that changes in the location of a line may occur due to expansion of the piping. Consequently, it is desirable to investigate the longer lines for the effect of thermal expansion on the pitch of a line. Proper cutting short of vertical pipe lines will avoid forming pockets which will collect condensate. Pitch of high pressure steam piping should be in the direction of steam flow if at all possible, and gravity drainage to the trap is desirable to secure positive drainage.

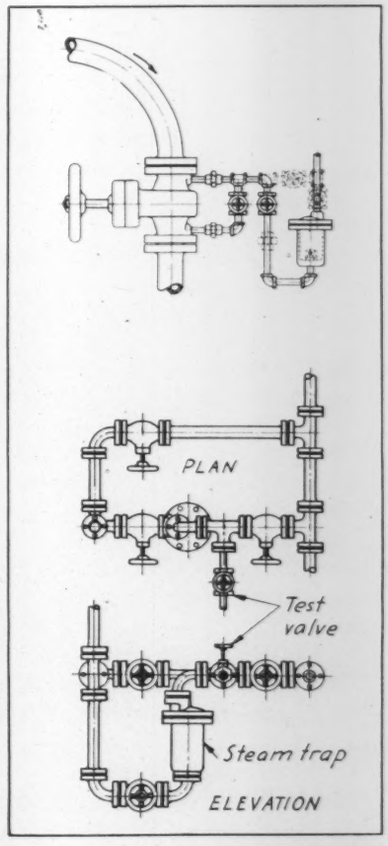
Selecting Trap Capacity

Traps should be selected in accordance with their water handling capacity rather than by the size of the inlet and outlet connections. The capacity of any trap in pounds of water discharged per hour depends upon the effective pressure at the trap inlet and the net area of the discharge valve opening. By "effective pressure" is meant the difference between the inlet pressure at the trap and the pressure in the discharge line from the trap, which is composed of the actual pressure existing in the discharge line plus the pressure due to the static head of water which may exist in this line.

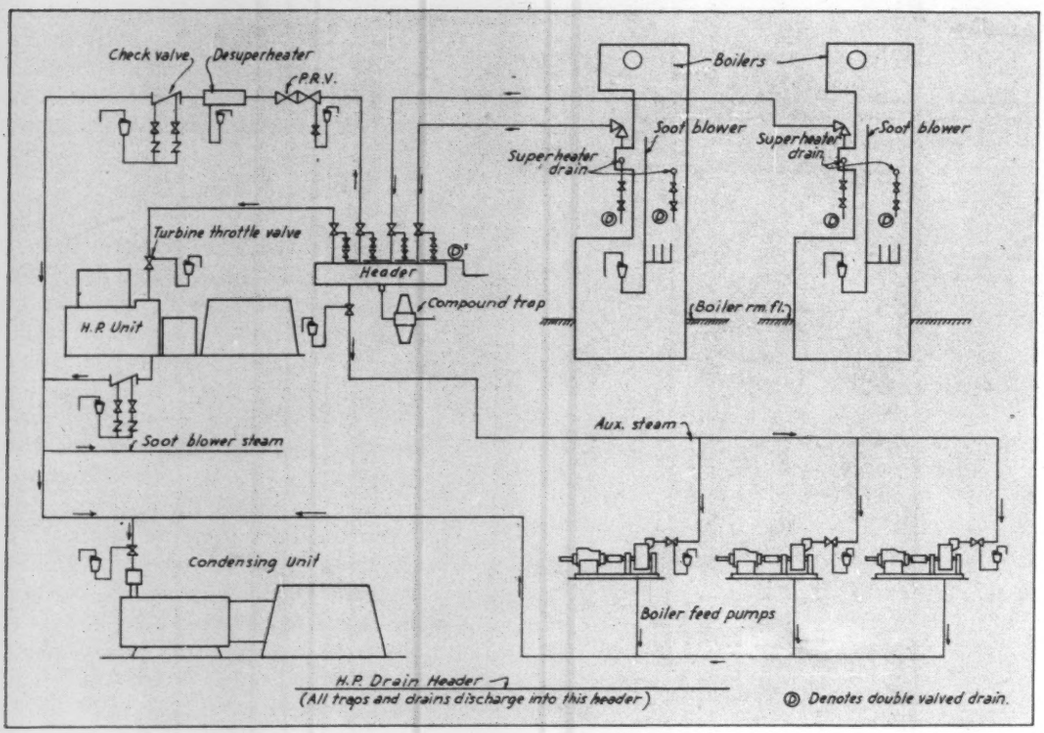
A single trap may be used to handle more than one drip, provided the drips are so grouped that they are at approximately the same pressure. Of course, valving in this case should be given careful consideration.

The following formula determines the amount of steam that is actually condensed in bringing the pipe from room temperature up to approximate steam temperature, and also reduces that quantity of steam to an hourly rate of flow:

C = Hourly rate of flow of condensate, lb;
W = total weight of pipe involved, lb;
T_2 = initial temperature of pipe, F;
T_1 = final temperature of pipe, F;



***Above*—Forged steel trap installed ahead of steam valve that is normally closed and operated frequently. *Below*—Flanged steel trap installed with bypass and test valve**



Schematic diagram showing typical practice for high pressure piping

L = latent heat of steam at final temperature;
0.114 = specific heat of steel pipe;

Heating up period: The time in minutes allowed to bring the pipe temperature up to steam temperature, which is left to the discretion of the engineer in charge, usually being about six to ten minutes.

$$C = \frac{W \times (T_1 - T_2) \times 0.114 \times 60}{L \times (\text{Heating up period})}$$

It is recommended that the trap to handle this warming up load should have a continuous discharge capacity somewhat in excess of that obtained by the above formula, a safety factor of 1½:1 being advisable.

Protection from Boiler Priming

The foregoing pertains only to the trapping of steam mains to protect the piping system in general, pressure reducing valves, throttle valves, and similar equipment from the regular and expected condensate load. In addition to this, it is often advisable to add a heavy duty or high capacity trap at the boiler to protect the equipment from the heavy surges due to boiler priming, or to drain separators which are installed where priming is encountered. The trap body, mechanism, valve and seat should follow the specifications given previously for drip traps but the trap should have capacities of 30,000 lb per hr and higher according to the steam generating capacity of the boiler. (As an example of capacities required for this service, an installation recently made in a central station required a 150,000 lb per hr trap capacity for a boiler capacity of 1,000,000 lb per hr. A 450,000 lb per hr industrial boiler installation required a 40,000 lb per hr trap capacity).

These traps are frequently of the compound type where a pilot trap discharges against a piston operated poppet valve giving practically a straight blow through for the condensate.

Piping Code Requirements

Perhaps the whole conception of the drainage system can be clarified by reference to the following extracts from the "Code for Pressure Piping":

118b. Drip lines from steam headers, mains, separators, and other equipment shall be properly drained with the traps installed in accessible locations and below the level of the apparatus drained, if practicable. All drain lines shall be provided with drip valves for free blow to the atmosphere.

118c. Drip lines from steam headers, mains, separators, or other equipment operating at different pressures shall not be connected to discharge through the same trap. Where two or more drips discharge into the same trap, a check valve and a stop valve shall be placed in each line. Where several traps discharge into one header under pressure, or which may be under pressure, a stop valve and a check valve shall be placed in the discharge line from each trap.

Double Valved Drains

Ordinarily when equipment is constantly operating and shutdowns are infrequent, traps are not provided for draining valves above the seats or at low points of the lines, but double valved drain lines discharging to the condensate return system are provided. These double valved drains are used only during the "heating up" period; after the system has reached the temperature where no more condensate is produced, the valves are closed and remain closed until such time when the system is reheated after a shutdown. The small amounts of heat recovery would not in these cases pay for the cost of trap installations and have practically no effect on the ultimate heat balance.

The double valving is necessary to insure positive closing of these drain lines, being used in the same manner and for the same purpose as blow-off valves.

The Manufacturers Standardization Society of the Valve and Fittings Industry, 420 Lexington Ave., New York, N. Y., has adopted Addendum No. 1 to SP-25-1936 which provides for standard marking of ring joint gaskets and flanges and also supplies a standard service symbol for air. This expands the application of the M. S. S. standard marking system for valves, fittings, flanges, and unions.

FOLLOWING are the appendices for the article "The Evaporative Condenser," by William Goodman, published in HEATING, PIPING AND AIR CONDITIONING, March, 1938, pp. 165-168, and April, 1938, pp. 255-258.

Appendix 1—Derivation of Equation 1

The total heat transferred from the water to the air, while the air is flowing past an infinitesimal area of wetted condenser surface, dA, is given by the following formula[1]:

$$dH = \frac{f_g}{0.24} [h_w - h] \, dA \quad \ldots\ldots [1]$$

where h is the enthalpy of the air at the instant it passes over the element of area dA. But

$$dH = G dh \quad \ldots\ldots [2]$$

Therefore,

$$G dh = \frac{f_g}{0.24} [h_w - h_a] \, dA \quad \ldots\ldots [3]$$

or,

$$\frac{dh}{h_w - h_a} = \frac{f_g}{0.24G} dA \quad \ldots\ldots [4]$$

Inasmuch as the temperature of the spray water is constant, h_w is also constant. Integrating equation [4] between the limits $h = h_1$ and h_2, and $A = 0$ and A yields

$$\text{Log}_e \frac{h_w - h_1}{h_w - h_2} = \frac{f_g A}{0.24G} \quad \ldots\ldots [5]$$

Let

$$\frac{f_g A}{0.24G} = M \quad \ldots\ldots [6]$$

Therefore,

$$\text{Log}_e \frac{h_w - h_1}{h_w - h_2} = M \quad \ldots\ldots [7]$$

But the change in enthalpy of the air flowing through the condenser is

$$H = G (h_1 - h_2) \quad \ldots\ldots [8]$$

Eliminating h_2 between equations [7] and [8], the following is obtained:

$$H = G (h_w - h_1) (1 - e^{-M}) \quad \ldots\ldots [9]$$

where e = base of natural logarithms.

Let

$$Z = (1 - e^{-M}) \quad \ldots\ldots [10]$$

Substituting [10] into [9] results in equation [1] of the text. Table 1 was prepared by means of equation [10] above.

[1]Goodman, W., *Refrigerating Engineering*, October, 1936, Appendix 1, p. 249.

Appendix 2—Derivation of Method for Determining Temperature of Spray Water by Means of Fig. 2 and Equation 3

Inasmuch as the temperature of the condensing refrigerant and the temperature of the spray water are theoretically constant throughout the condenser, the temperature of the metal surface of the coil must also be constant throughout. The temperature of the metal surface must be at a point between the temperature of the condensing vapor and the temperature of the spray water.

The heat transferred from the condensing refrigerant to the wall of the metal tube is

$$H = f_R (t_R - t_{s1}) \frac{A}{B} \quad \ldots\ldots [1]$$

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The Evaporative Condenser

By **William Goodman***

and

$$H = f_w (t_{so} - t_w) A \quad \ldots\ldots [2]$$

where

t_{s1} = temperature of the inside surface of the metal tube;
t_{so} = temperature of the outside surface of the metal tube.

Inasmuch as the metal wall of the tube is thin and its conductivity comparatively high, the resistance of the metal wall to the flow of heat is negligible. Hence, the temperatures of the inside and outside surfaces of the metal will be substantially equal. Therefore, if $t_{s1} = t_{so}$, they may be eliminated between [1] and [2], yielding

$$H \left[\frac{1}{f_w} + \frac{B}{f_R} \right] = (t_R - t_w) A \quad \ldots\ldots [3]$$

Let

$$\frac{1}{U} = \frac{1}{f_w} + \frac{B}{f_R} \quad \ldots\ldots [4]$$

Substituting [4] into [3] results in

$$H = U (t_R - t_w) A \quad \ldots\ldots [5]$$

which is equation [4] of the text.

Dividing equation [1] of the text by [5] as given above yields

$$1 = \frac{G (h_w - h_1) Z}{U (t_R - t_w) A} \quad \ldots\ldots [6]$$

or,

$$\frac{t_R - t_w}{h_w - h_1} = \frac{GZ}{AU} \quad \ldots\ldots [7]$$

Let

$$N = \frac{GZ}{UA} \quad \ldots\ldots [8]$$

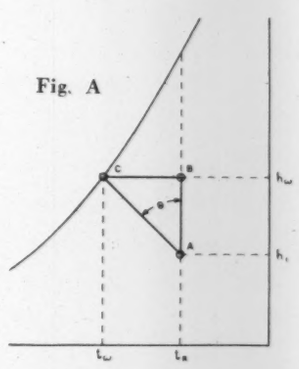
This is equation [3] of the text.

Therefore,

$$\frac{t_R - t_w}{h_w - h_1} = N \quad \ldots\ldots [9]$$

For any constant value of N, this is the equation of a straight line on a chart whose abscissa is water temperature and whose ordinate is the enthalpy of the air. Fig. 2 was prepared in this way. The enthalpy values are not shown on Fig. 2; instead, values of wet bulb temperature corresponding to the enthalpy of saturated air were used for simplicity in solving problems. The curve in Fig. 2 was obtained by plotting values of the enthalpy of saturated air against dry bulb temperature. Referring to the skeleton chart in Fig. A,

Fig. A



$$\frac{CB}{AB} = \tan \theta \quad \ldots\ldots [10]$$

Length of line CB is,

$$CB = s_A (t_R - t_w) \quad \ldots\ldots [11]$$

Similarly, length of line AB is,

$$AB = s_o\ (h_w - h_1) \quad \ldots\ldots [12]$$

where

s_A = scale of abscissa, inches per degree Fahrenheit;
s_o = scale of ordinate, inches per Btu.

Dividing [11] by [12],

$$\frac{CB}{AB} = \frac{s_A\ (t_R - t_w)}{s_o\ (h_w - h_1)} \quad \ldots\ldots [13]$$

Substituting [10] into [13],

$$\frac{t_R - t_w}{h_w - h_1} = \frac{s_o}{s_A} \tan\theta \quad \ldots\ldots [14]$$

Fig. 2 was so drawn that $\frac{s_o}{s_A} = 1$.

Equating [9] and [14],

$$N = \tan\theta \quad \ldots\ldots [15]$$

Table 2 was prepared by means of this equation.

Appendix 3—Derivation of Equation 6

As is evident in Fig. 6, the temperature of the spray water becomes less as G increases; it approaches a limiting value when G = infinity. Equation [3] cannot be used to find the angle θ when G = infinity, because for this condition equation [3] reduces to $N = \infty \cdot 0$, which is indeterminate. Therefore, it is necessary to find the equation of N for the condition when G = infinity.

Equation [3] of the text is

$$N = \frac{GZ}{UA} \quad \ldots\ldots [1]$$

Substituting equations [6] and [10] of appendix 1, equation [1] can be written in the following form:

$$N = \frac{\left[1 - e^{-\frac{f_g A}{0.24G}}\right]}{\frac{UA}{G}} \quad \ldots\ldots [2]$$

The limiting value of N when G = infinity is found by the well-known method of differentiating the numerator and denominator separately with respect to G, yielding

$$N = \frac{\left[\frac{f_g A}{0.24}\right]\left[e^{-\frac{f_g A}{0.24G}}\right] d\left[\frac{1}{G}\right]}{UA\ d\left[\frac{1}{G}\right]} \quad \ldots\ldots [3]$$

The factor $d\left[\frac{1}{G}\right]$ is eliminated by cancellation, as it appears in both the numerator and denominator.

When G = infinity and $N = N_L$, [3] above reduces to

$$N_L = \frac{f_g}{0.24U} \quad \ldots\ldots [4]$$

This is equation [6] of the text.

Appendix 4—Proof that the Condition of the Air Flowing Through an Evaporative Condenser Can Be Represented by a Straight Line on a Psychrometric Chart

The total weight of moisture evaporated from the spray water while the air is flowing past an infinitesimal area of wetted condenser surface, dA, is given by the following formula[2]:

$$G\,dw = \frac{f_g}{0.24}\ [w_w - w]\ dA \quad \ldots\ldots [1]$$

or,

$$\frac{dw}{w_w - w} = \frac{f_g}{0.24G}\ dA \quad \ldots\ldots [2]$$

where

w = absolute humidity of the air at the instant it passes over the element of area, dA.
w_w = absolute humidity of saturated air at a temperature equal to the temperature of the spray water.

Inasmuch as the temperature of the spray water is constant, w_w is also constant. Integrating [2] between the limits $w = w_1$ and w_2, and $A = 0$ and A yields,

$$\text{Log}_e \frac{w_w - w_1}{w_w - w_2} = \frac{f_g A}{0.24G} \quad \ldots\ldots [3]$$

Equating this equation to [5] of appendix 1 and taking antilogarithms, yields

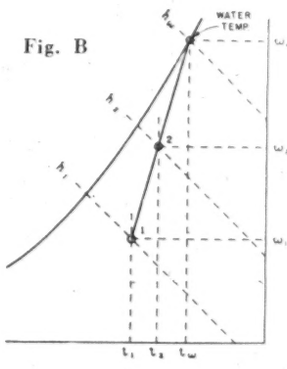
$$\frac{w_w - w_1}{w_w - w_2} = \frac{h_w - h_1}{h_w - h_2} \quad \ldots\ldots [4]$$

or

$$\frac{w_w - w_1}{h_w - h_1} = \frac{w_w - w_2}{h_w - h_2} \quad \ldots\ldots [5]$$

This is evidently the equation of a straight line on a psychrometric chart whose coordinates are enthalpy and absolute humidity, as in Fig. B. It makes no difference whether the angle between the ordinate and the abscissa is a right angle or an acute angle. Inclining the enthalpy lines as in Fig. B results in a chart that has the appearance of the familiar psychrometric chart with the lines of dry bulb temperature almost vertical.

Fig. B



[2]Keevil and Lewis, *Industrial and Engineering Chemistry*, October, 1928, p. 1058.

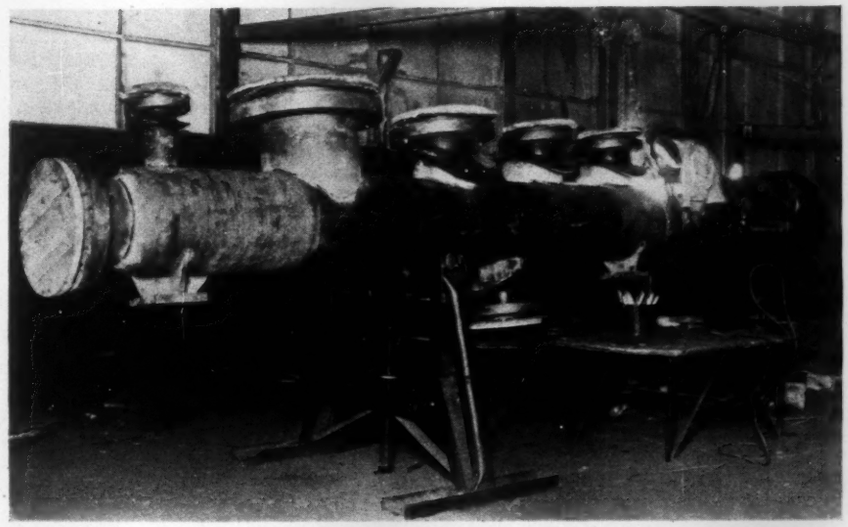
Fuel Management Program for Schools

["A Critical Analysis of the Fuel Management Program for Schools," by Ashley M. McCullough. 1st ed. 1937. 141 + viii pp., 6 x 9 in., clothbound. Published by the Bureau of Publications, Teachers College, Columbia University, New York, N. Y. Price, $1.85.]

This publication reviews in detail the problem of purchasing fuel for schools and managing its use—a subject of prime importance to school administrations, for "the annual purchase of fuel entails the largest expenditure for a single supply made by the school system of the nation."

In his introduction, the author discusses previous studies concerning fuel, sets up the problem, and outlines the purpose and scope of his study. He then proceeds with chapters describing the methods of investigation used, validity of the basic requirements, an analysis of the fuel management program in eight selected New Jersey cities, and the present status of the fuel management program in certain U. S. cities. He then presents a fuel management program for any school system and concludes with a chapter on applications and recommendations.

A useful bibliography of 4½ pages is included.



Shop prefabrication of large header showing welding of branch connections. The work is supported by the rollers in the foreground and revolved by a machine so that the weld is made in a downward position, the operator controlling the revolving machine by a foot operated switch. Note that the saddles are slipped up on the branch pipes. The saddle for the large opening is left off as it would interfere with the pressure weld. It is put on later in two sections and welded

Design and Fabrication of High Pressure High Temperature Welded Piping

F. C. Fantz* discusses here the various types of welds used in the fabrication of high pressure and high temperature piping, and gives special attention to the design of backing up rings, and branch and intersection welds . . . This information has been taken from the paper presented by Mr. Fantz at the recent Western Regional Conference of the American Welding Society

THE welding used in a piping system can be divided into three general classes, butt welds, fillet welds, and branch or intersection welds. The latter type is a combination of the butt and fillet welds.

Butt Welds

The butt weld is the easiest form of weld to make and at the same time produce a joint with the same factor of safety as the parts joined. The ends to be welded should preferably be machined in order to produce a smooth and uniform welding groove. When cut with a gas cutting torch, the surfaces should be ground or otherwise smoothed to prevent slag inclusions and lack of fusion in the finished weld.

The various wall thicknesses should be prepared as shown in Fig. 1.

High pressure lines should be prefabricated in a shop in units that can be shipped and handled, and all welds to be made at the site of erection should be limited to butt welds, preferably with the pipe in a horizontal position. Advantages of shop prefabrication are due to the trained personnel available, equipment especially designed for pipe fabrication, and the use of special machines and welding operators continuously employed for high pressure fabrication.

Inasmuch as most welds made at the site cannot be cleaned out after welding, it is thought that some form of backing ring is necessary to prevent the introduction into the lines of small beads of metal. These beads adhere to the inside of the pipe for a time and later loosen up and may cause serious damage to valves, turbine blading and other equipment. Designs of high pressure butt joints using backing rings are shown in Figs. 2-8.

In shop fabrication this is not necessary, as the various assemblies can be cleaned out with machine cleaning wheels. Turbines are now in use for pipes of diameters up to 20 in. and can pass around a curve with a radius equal to three times the diameter of the pipe. Pipe units that have been stress relieved are cleaned by this method to remove tight scale and foreign matter that may be burned to the pipe wall during the bending or stress relieving operations.

Backing Up Rings

Our experience, covering a period of over 15 years of research in high pressure welded connections, has led us to the following conclusions concerning backing up rings:

They should not be heavy enough to cause chilling of the first layer of deposited metal.

The inside diameter of the pipe and the ring should be machined to a close fit. This will prevent arc blow and lack of fusion so common when the ring does not fit close.

The ring and pipe end should be tapered to insure easy assembly and to center the ring in the joint; also to compensate for variations in inside diameter of mating sections.

The ring should be solid and not split.

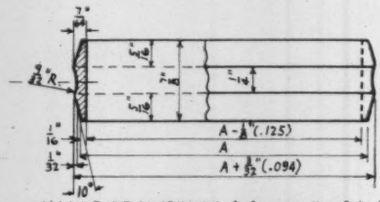
The inside diameter of pipe should not be restricted more than 2 per cent.

Thorough fusion must be obtained between the ends of the pipe and ring.

*Vice-President, Midwest Piping and Supply Co., Inc.

TABLE 1

BACKING RING FOR BUTT WELDS IN HIGH PRESSURE & HIGH TEMPERATURE SERVICE
Recommended for Wall Thicknesses .500" & Greater



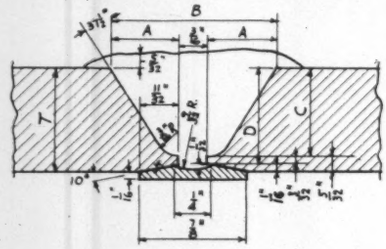
VALUE OF DIMENSION A (Inside dia. of pipe)

PIPE SIZE	Schedule No. For Wall Thicknesses (A 106 - 36)							
	30	40	60	80	100	120	140	160
4							3.438	3.438
5					4.563	4.563	4.313	4.313
6					5.501	5.501	5.189	5.189
8				7.625	7.439	7.189	7.000	6.813
10				9.750	9.564	9.314	9.064	8.750	8.500
12				11.626	11.376	11.064	10.750	10.500	10.126
14				12.814	12.500	12.126	11.876	11.500	11.188
16			15.000	14.688	14.314	13.938	13.564	13.126	12.876
18			16.876	16.564	16.126	15.688	15.314	14.876	14.500
20	19.000		18.814	18.376	17.938	17.500	17.000	16.500	16.126
24	22.876		22.626	22.126	21.564	21.000	20.500	19.786	19.376

All dimensions are in inches.

TABLE 2

BACKING RING FOR BUTT WELDS IN HIGH PRESSURE & HIGH TEMPERATURE SERVICE
Recommended for Wall Thicknesses .500" & Greater

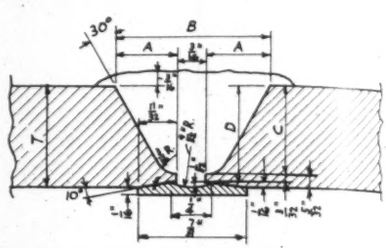


PIPE WALL T	A	B	C	D
.531	.443	1.073	.375	.468
.562	.467	1.121	.406	.500
.593	.491	1.169	.437	.530
.628	.518	1.223	.472	.565
.643	.529	1.235	.487	.580
.656	.539	1.265	.500	.593
.687	.563	1.313	.531	.624
.718	.587	1.361	.562	.655

All dimensions are in inches.

TABLE 3

BACKING RINGS FOR BUTT WELDS IN HIGH PRESSURE & HIGH TEMPERATURE SERVICE
Recommended for Wall Thicknesses .500" & Greater



PIPE WALL T	A	B	C	D	PIPE WALL T	A	B	C	D
.750	.437	1.060	.594	.6875	1.156	.671	1.530	1.000	1.093
.812	.472	1.132	.656	.7495	1.218	.707	1.602	1.062	1.155
.843	.490	1.168	.687	.780	1.250	.725	1.638	1.094	1.187
.906	.527	1.242	.750	.843	1.312	.761	1.710	1.156	1.249
.937	.545	1.278	.781	.874	1.343	.779	1.745	1.187	1.280
1.000	.582	1.351	.844	.937	1.406	.815	1.818	1.250	1.343
1.031	.599	1.386	.875	.968	1.437	.823	1.834	1.281	1.374
1.062	.614	1.416	.906	1.000	1.500	.870	1.927	1.344	1.4375
1.125	.653	1.494	.969	1.063					

All dimensions are in inches.

Full strength based on pipe wall thickness must be maintained after welding.

The preparation of the ring and tube ends should not be expensive, and should not require special machines to produce.

Tables 1-3 show the design and dimensions of ring and pipe ends that we have tested and found to comply with the above requirements.

The ring is grooved in the center to a width of ¼ in. and 1/32 in. deep. This allows the 1/16 in. wide lip to extend over the groove 1/32 in. and enables the welding operator to make perfect fusion of the pipe ends and ring without using excessive heat with the first layer.

This design of backing is a result of research and experimental work and is considered an improvement over the one shown in Fig. 8, as no upsetting or flaring is required on the pipe ends.

Fillet Welds

This type of weld is used in attaching slip-on and screwed flanges. The A.S.A. Code for Pressure Piping does not recommend slip-on flanges for over 300 lb working pressure. Some of the oil refining companies are using this type of flange successfully for pressures as high as 900 lb. However, it is thought that the butt welding flange can be applied with less distortion of the face and with less expense, all things considered.

On small pipes the fillet weld is used as shown in Fig. 9. The fillet weld is also used in attaching pedestals and supports for headers and elbows, anchors to pipe lines, and various reinforcing members for branch intersections.

Branch and Intersection Welds

The header or manifold of today is a combination of tubing and various welding fittings prefabricated by welding in the shop. In some cases the valves are welded direct to the openings during fabrication and then made a part of the piping system by welding at the construction site. The design and execution of this type of weld is much more difficult than either the butt or fillet welds and should not be attempted in the field.

The three methods of making branch connections having the widest use are shown in Figs. 10-13.

The example of Fig. 11 is prepared for welding by inserting the branch flush with the inside diameter of the header. The end of the branch changes from a square cut at point *A* to a 60 deg bevel at point *B*. The hole in the header is also beveled.

Fig. 12 shows that both the branch end and the hole in the header are beveled. Both methods require very careful layout and fitting in order to produce the proper welding groove.

What is thought to be a better method of preparation for a branch connection is shown by Fig. 13. In all cases when this method is used, the hole in the header is cut to dimensions of the inside diameter of the branch and parallel to the side walls of the branch. This does not require an elaborate technique. The holes for small branches can be drilled or bored, the larger sizes cut with the torch (either manual or machine guided). Considerably less material is removed from the header.

The preparation of the branch pipe end is a true development of the intersection of cylinders and can be laid out on a template and transferred to the work or cut by a machine guided torch capable of developing this intersection. Such machines have been perfected and are now on the market. When the hand manipulated torch is used, it is held at a constant angle with relation to the axis of the pipe. This manipulation is not as difficult as in the methods previously described and can easily be handled by the average welding operator.

It should be remembered that the easier it is made for the operator to prepare the connection and deposit the weld metal, the greater assurance there is of obtaining uniform good quality welds.

When a section is cut out of a pipe and a branch is inserted, the strength of the original pipe is impaired to a degree depending upon the relation of size of branch to header, and the relation of wall thickness to the diameter of the header section. In the design of relatively short headers with many branches, it is common practice to make the header wall heavier than would be necessary if

no holes were cut, thus eliminating the need for additional reinforcing.

When the wall thickness is based on pressure and temperature conditions, some form of reinforcing the branch connection is necessary. Where the branch pipe diameter is one-third, or less, than the diameter of the header, it is thought that the addition of weld metal properly applied would act as a reinforcing ring. As the size of the branch increases in relation to the diameter of the header, additional reinforcing is required.

As the design approaches size on size (12 in. branch to a 14 in. header), the flat portion at *B*, Fig. 10, must be considered. The increased cross-sectional area at this point concentrates a greater strain at point *A;* and if pressure is raised to the bursting point, rupture will always occur at this point. The use of gussets or flat ribs between the branch and header have not proved satisfactory and have been replaced with a ring or saddle.

Above **— Example of branch connections reinforced with saddles. The connection in the side of the welding ell has a saddle shaped to fit the ell. . . .** ***Below*** **— Method of providing a base for a 12 in. ell and riser pipe**

The ring type can be made without the use of dies and can be cut and rolled or shaped to the header in the field. For this reason the ring type

Fig. 1—Methods of preparing various wall thicknesses for butt welding

Fig. 2—High pressure butt joint using backing ring. *T* = pipe wall thickness. Upset generally *T* plus 25 per cent. Note spacing at *A* to allow shrinkage of weld. Easily assembled due to tapered ring. Full inside diameter maintained. Expensive to upset and machine

Fig. 3—High pressure butt joint using backing ring. Pipe ends not upset. Full strength maintained. Difficult to assemble due to straight ring and non-uniform inside diameter of pipe. Flow restricted by reduced inside diameter

Fig. 4—High pressure butt joint using backing ring. Full strength maintained. Difficult to assemble. Difficult to make close fit between pipe and ring

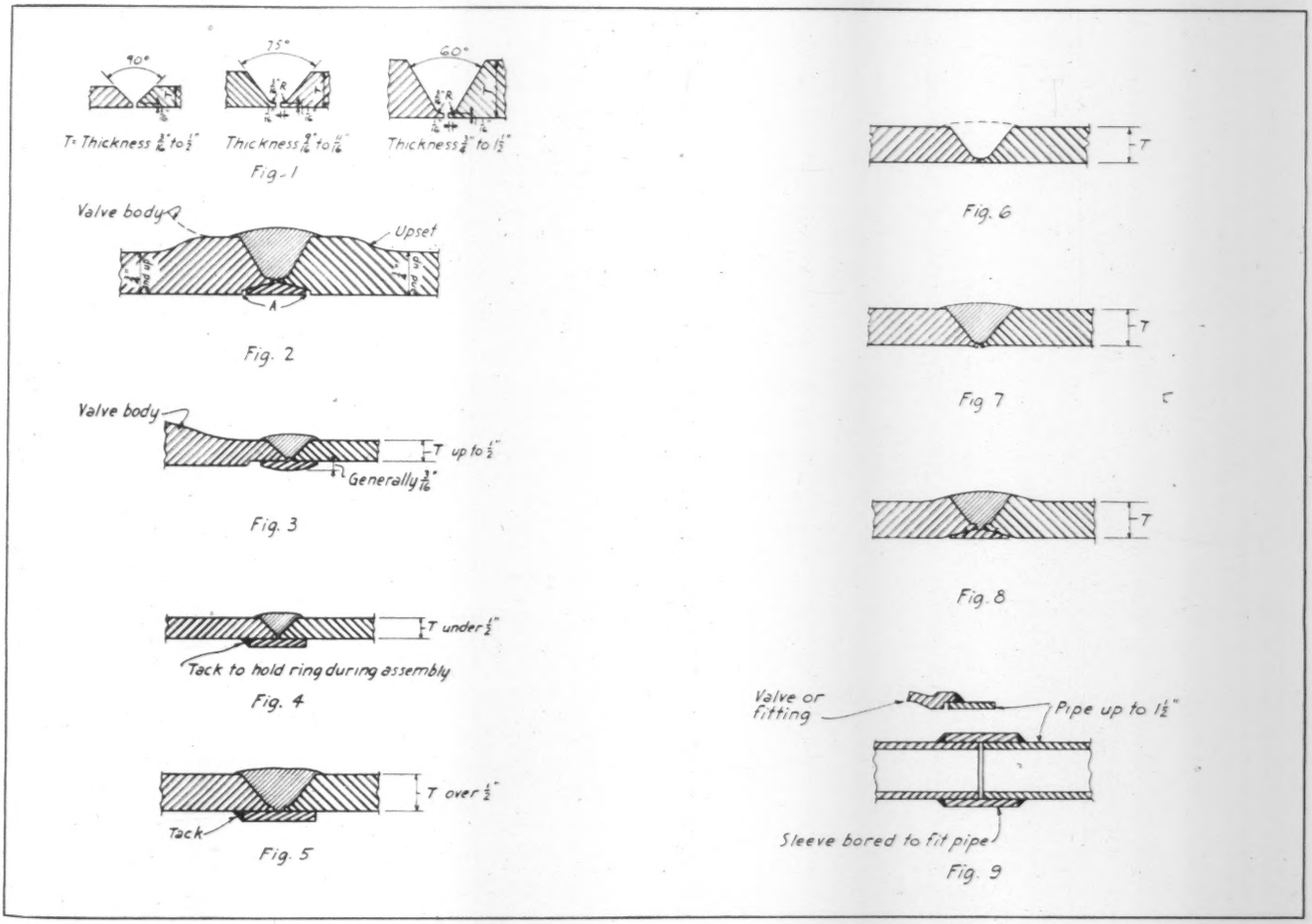
Fig. 5—High pressure butt joint using backing ring. Full strength maintained. Difficult to assemble. Difficult to make close fit between pipe and ring

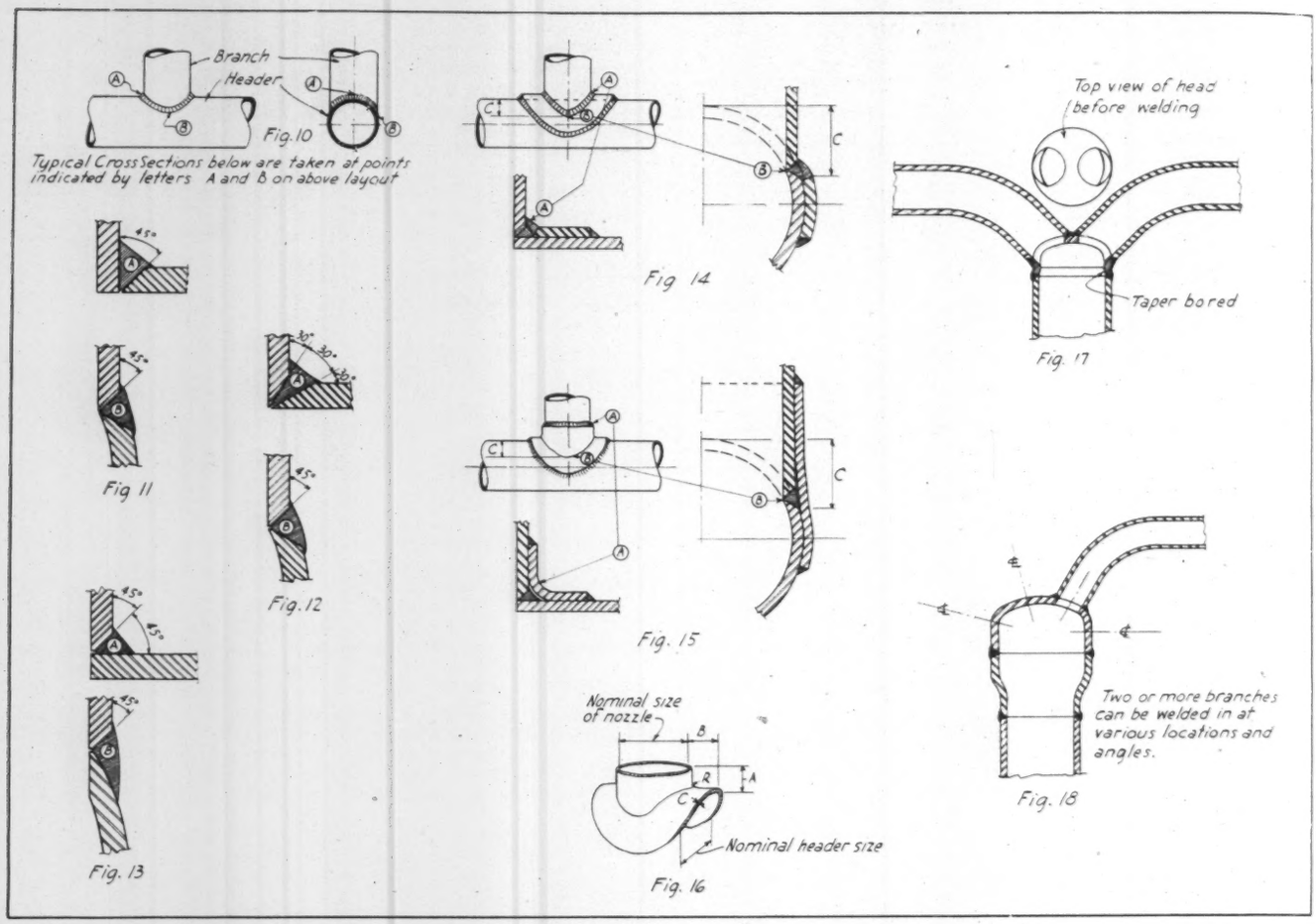
Fig. 6—No backing ring. No space between pipe ends before welding. Full strength maintained. Difficult to line up. Requires extreme care in depositing first layer of weld

Fig. 7—Note upset of pipe ends to inside. Appearance of Fig. 6 after welding. Slight opening at inside not considered detrimental, as it is not a crack with sharp edges and has very little chance of causing progressive fracture

Fig. 8—Note groove cut in ring and extension of pipe ends over groove. Ends of pipe flared and upset. Full strength maintained. Easy to line up and assemble. Easy to obtain thorough fusion at bottom of groove and ring. Flare and upset is expensive to make and machine, although less than that of Fig. 2

Fig. 9—Fillet weld for small pipes





Figs. 10, 11, 12 and 13—Methods of making branch connections
Fig. 14—Ring plate type reinforcement. Note that flat portion *C* is not reinforced
Fig. 15—Saddle type reinforcement. Flat portion *C* is reinforced by the ring section of the saddle which encircles the branch
Fig. 16—Reinforcing saddle sizes. See Table 4 for dimensions
Figs. 17 and 18—Branch connections made by welding two or more branches into a welding head

Table 4—Reinforcing Saddle Sizes

Nominal Size of Nozzle	Nominal Header Sizes	See Fig. 16			
		A	B	C	R
2	2 to 24	1 1/4	1 3/4	1/2	1/2
2 1/2	2 1/2 to 24	1 1/2	1 3/4	1/2	1/2
3	3 to 24	1 3/4	2	1/2	1/2
3 1/2	3 1/2 to 24	1 3/4	2	1/2	1/2
4	4 to 24	2	2	1/2	3/4
5	5 to 24	2	2	1/2	3/4
6	6 to 24	2 1/4	2 1/2	1/2	3/4
8	8 to 24	3	3	5/8	1
10	10 to 24	3 1/2	3 1/2	5/8	1
12	12 to 24	4	4	5/8	1 1/4
14 O.D.	14 to 24	4	4	5/8	1 1/2
16 O.D.	16 to 24	4	4	5/8	1 1/2
18 O.D.	18 to 24	4	4	5/8	1 1/2
20 O.D.	20 to 24	5	5	5/8	1 1/2
24 O.D.	24	5	5	5/8	1 1/2

is advocated by some engineers. However, reinforcing saddles can be obtained from stock of several manufacturers in the sizes given in Table 4. The cost of the finished assembly will compare very favorably with the ring plate type.

The ring portion of the saddle not only supports the flat section of the connection, but it also tends to relieve the weld of concentrated strains set up by movement of the connecting branch line caused by expansion and contraction, or so-called "hammer shock."

Where the ring or saddle is applied to assemblies which are to be later stress relieved or annealed, a small hole should be drilled through the reinforcing member to relieve pressure that may build up due to expansion of entrapped air or gas. Serious deformations have been observed in large diameter pipes when this precaution was not taken.

Another design of branch connection that is proving quite successful for high pressure and temperatures is made by welding two or more branches into a welding head. The head should be of heavier wall material to compensate for the reduction in strength of the openings. Two examples are shown in Figs. 17 and 18.

The author fully realizes that there may be other designs and methods of welding that will meet the desired requirements, and this paper is given with the hope that it will at least bring out some new points and raise some questions that will in the long run help to advance the art of welding.

Building Holds Key to Prosperity

The building industry holds the key to permanent prosperity, and the accumulated demand for new construction is a challenge to the advertising and sales brains of America, James G. Caffrey, president of Build America, told the New York Executives Club at the Biltmore Hotel last month. "Private construction, privately financed," Mr. Caffrey said, "is essential to the rehabilitation of the industry, and the success and rapidity with which this end is realized will be measured by the intelligent effort of manufacturers and distributors and the cooperation they receive from governmental agencies.

Heating · Piping and Air Conditioning

American Society of Heating and Ventilating Engineers

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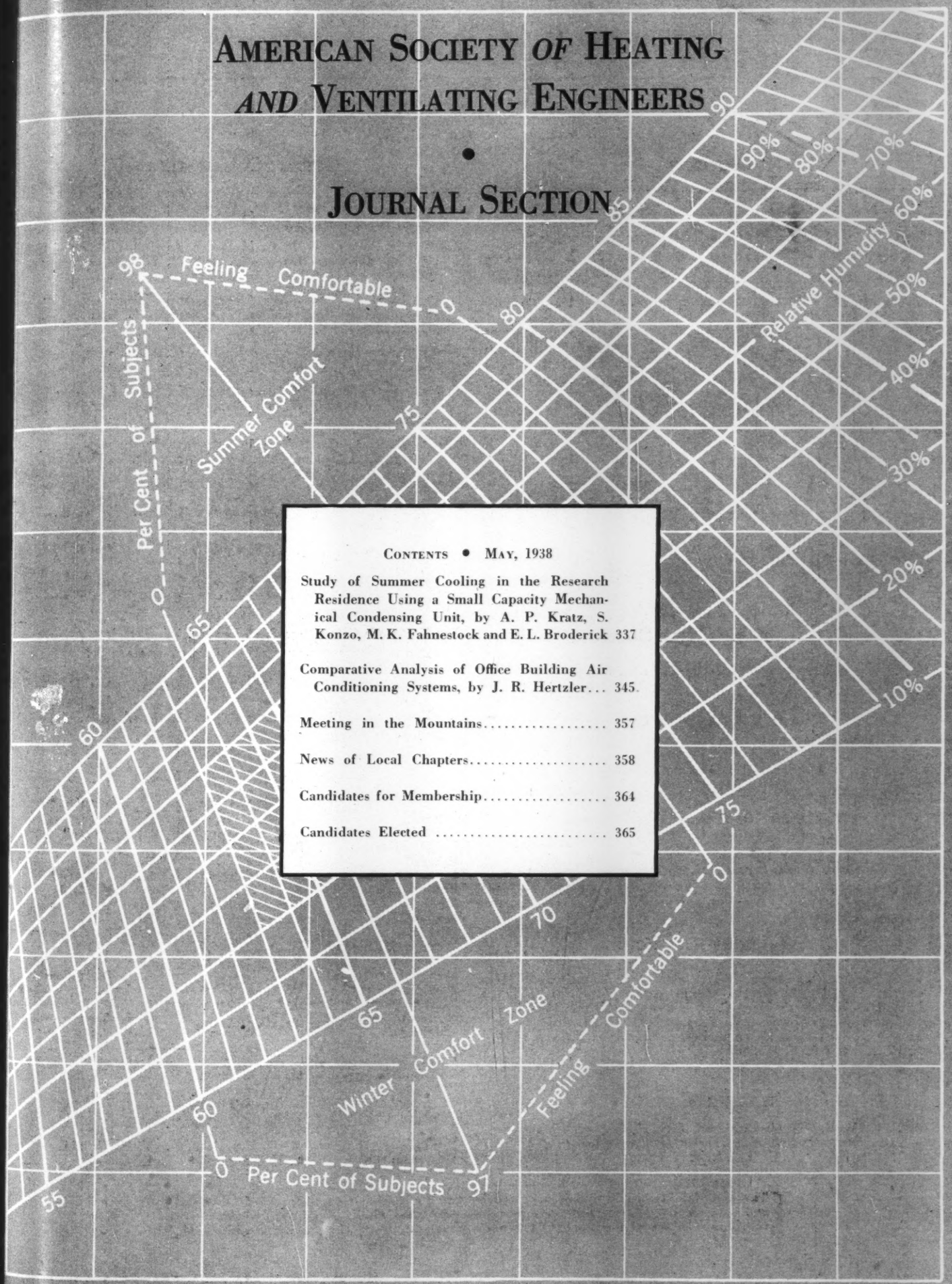
Journal Section



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Semi-Annual Meeting 1938

All members are invited to attend the Summer Meeting at The Homestead, Hot Springs, Va., June 20-22. **See program details page 357.**

* * *

Chicago Adopts Society Standards

The Code of Minimum Requirements for Comfort Air Conditioning prepared by the joint committee of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *American Society of Refrigerating Engineers* has been adopted by the general advisory group of the Chicago Committee on Air Conditioning in lieu of the Standards previously prepared by them.

It was agreed at a meeting of the Technical Committee of this organization on April 14, 1938 that the code shall be reprinted and distributed for general Chicago use, including an explanatory supplement cross-indexed to THE GUIDE and condensed on one sheet for insertion in the code. The supplement contains additional requirements applying to local conditions. It is the intention to revise and reprint this supplement every year.

* * *

Members' Cooperation Requested

The attention of members is called to the provisions of the Society's By-Laws which relate to the endorsement of Candidates for membership. Those who sign membership applications as Proposers and Seconders should be personally acquainted with the Candidate and should have a thorough knowledge of his engineering ability. The Committee on Admission and Advancement requires authoritative data from those with whom he is closely associated in his work, and it is obviously unfair to the applicant, to the Society, and to the endorsers if slightly or totally unknown individuals are proposed for membership.

The applicant for membership *seeks* admission into the Society and the grade of membership is *conferred* upon him in accordance with his qualifications, and the Committee on Admission and Advancement can only base its judgment upon the written statements submitted by the Candidate and the supporting data of his Proposers and Seconders.

The cooperation of all members will be appreciated by the Committee on Admission and Advancement to advance the growth and prestige of the A.S.H.V.E.

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Study of Summer Cooling in the Research Residence Using a Small Capacity Mechanical Condensing Unit

(Part 1)

By A. P. Kratz*, S. Konzo**, M. K. Fahnestock***, and E. L. Broderick† (*MEMBERS*)
Urbana, Ill.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the *National Warm Air Heating* and *Air Conditioning Association* and the University of Illinois.

THE previous investigations in summer cooling in the Research Residence during the summers of 1932 to 1936 inclusive made use of central cooling plants of approximately 30,000 Btu per hour capacity, employing either ice, mechanical refrigeration, or cold water. The investigation for the summer of 1937 was undertaken to determine the temperature and humidity conditions that could be maintained in the Research Residence with a mechanical condensing unit having a capacity approximately 40 per cent less than the 30,000 Btu per hour previously used; particularly when such cooling was supplemented by the circulation of outdoor air through the house at night, and when approximately one air change per hour of outdoor air was used for the purpose of ventilation during the day.

In the first series of tests (Series 1-37) made during the summer of 1937, cooled air was distributed to the rooms on both the first and second stories when the indoor air conditions were such that cooling was required. However, in the second series of tests (Series 2-37) the cooled air was delivered only to the rooms on the first story during the daytime, and only to the rooms on the second story after 6:30 p. m.

Description of the Research Residence and Cooling Equipment

The Research Residence together with the forced-air heating system has been described in a previous paper.[1] For the purpose of this investigation the Residence was equipped with awnings at all east, south, and west windows, and the sunroom was isolated from the rest of the house by closing the doors leading into the dining room. The entire third story was regarded as an attic and during the daytime was isolated from the rest of the house by closing the door at the head of the stairs. The attic windows, however, were opened to provide ventilation in the attic both day and night. The total space cooled amounted to 14,170 cu ft. Unless otherwise specified, the state of the Residence was the same as that during previous summers.

With the exception that the duct leading to the kitchen register was connected to the south trunk duct which supplied air to the first story rooms only, the arrangement of the forced-air duct system and fan was practically the same as that used in previous summers. For the first series of tests (Series 1-37), in which both the first and second story rooms were cooled at the same time, the dampers in the ducts to the first and second stories were adjusted to maintain the proper balance between the amounts of air required for cooling on these two stories. For the second series of tests (Series 2-37), in which the first story rooms were cooled until 6:30 p. m., and the second story rooms were cooled after 6:30 p. m., a damper arrangement was installed in the bonnet of the furnace casing so that the circulating air could be distributed either entirely through the registers on the first story or entirely through the registers on the second story.

The arrangement of the cooling plant is shown in Fig. 1. The condensing unit was self contained and consisted of a water-cooled shell and finned coil condenser-receiver, and a two-cylinder motor-driven compressor. Cooling was accomplished by direct expansion of dichlorodifluoromethane in an evaporator placed in a by-pass in the central cold air return duct. The evaporator, or cooling coil, consisted of finned copper tubing placed two rows deep in the direction of air flow and nine rows high measured along the vertical axis of the air duct. The tubes were ¾ in. outside diameter and had fins, eight to the inch, which were continuous strips of sheet metal extending from the top row to the bottom row of tubes. The evaporator as installed in the duct was 14⅝ in. high and 20⅜ in. wide, and the gross, or face, area presented to the flow of air was 2.07 sq ft, while the net, or free, area was 1.02 sq ft. The nominal capacity of the condenser and evaporator units under the conditions of actual operation at a compressor speed of 255 rpm, with return refrigerant temperature of 55 F, inlet water temperature of 59 F, ambient air temperature of 74 F, and with an air velocity of approximately 479 fpm across the 2.07 sq ft of face area, was 16,300 Btu per hour.

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[1] Study of Summer Cooling in the Research Residence at the University of Illinois, by A. P. Kratz and S. Konzo (A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 95).

For presentation at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Hot Springs, Va., June, 1938.

The controls employed consisted of a thermostatic expansion valve having the thermostatic bulb clamped to the suction line near the junction with the evaporator; a pressure operated water control valve, used to regulate the flow of water through the condenser; and a room thermostat, located in the hall on the second story. The latter served to start and stop the compressor in accordance with the cooling load required to maintain constant room temperature.

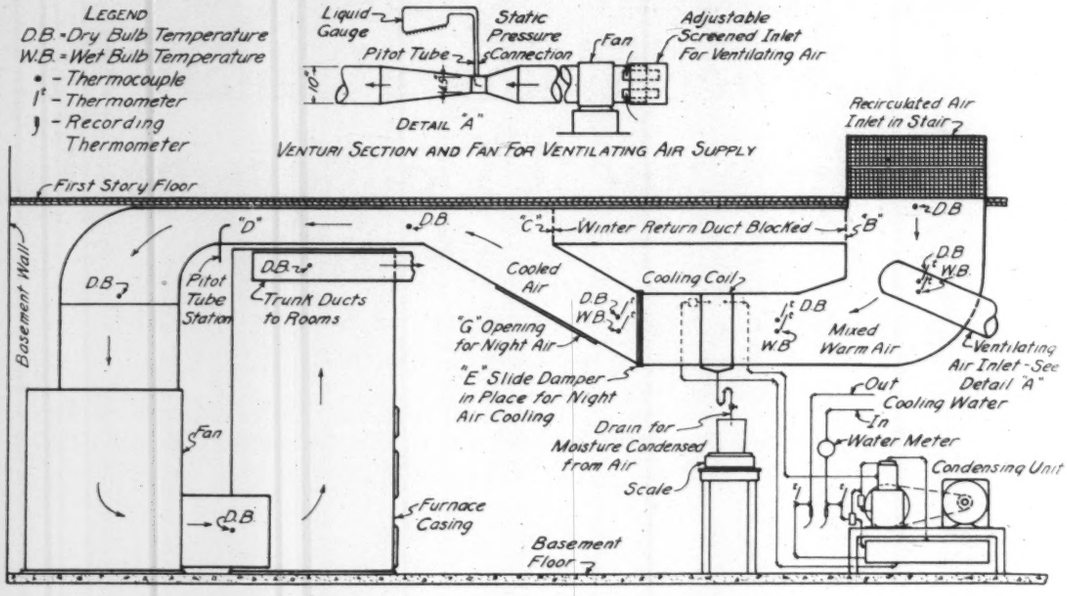

Fig. 1—Diagram of cooling plant with mechanical refrigeration

The cooling coil was installed in a by-pass in the central return duct, as shown in Fig. 1. For the summer work, the duct was blocked at B and C, and all of the air delivered to the fan in the forced-air system passed through the cooling coil when the air in the house was being recirculated. For the purpose of providing outdoor air for cooling during the night, a slide damper, E, was placed in the by-pass on the outlet side of the coil, and a door, G, was placed in the recirculating duct on the down-stream side of the slide damper. When outdoor air was required, the basement door and the door, G, in the recirculating duct were opened, and the slide damper, E, was closed. The fan in the forced-air system delivered approximately 1000 cfm of air when recirculating the air in the house and 2220 cfm when using outdoor air at night. These air volumes were based on an air density of 0.075 lb per cubic foot. Outdoor air, for the purpose of ventilation during the day, was provided by means of the duct shown as Detail A in Fig. 1.

The study of the heat flow through exposed west and north walls was made in the northwest bedroom of the Research Residence. Two Nicholls heat-flow meters[2] were installed, one on the west wall and one on the north wall, with the centers at an approximate height of 33 in. above the floor. The wall section consists of lap siding, building paper, sheathing, studding space, wood lath and plaster with rough sand finish. The coefficient of heat transmission for this uninsulated wall section is 0.25 Btu per square foot per hour per degree Fahrenheit, at a wind velocity of 15 mph. A thin felt pad was placed over the plaster wall and the heat-flow meter was fitted tightly against the felt pad by means of a wooden frame, as shown in Fig. 2. A complete set of 17 thermocouples, in addition to those incorporated in the heat-flow meters, was installed so that air temperatures and surface temperatures both underneath and adjacent to the heat-flow meters could be obtained.

Method of Conducting Tests

Continuous records were made during the summer of air temperatures in the various parts of the house and in the different sections of the duct system. A complete description of the measurements which were made has been given in a previous report.[3]

During the periods of operation, observations were made of the weight of water circulated through the condenser, the temperatures of the water entering and leaving both the condensing unit and the house, the head and suction pressures, and the electrical inputs to the compressor and fan motors.

During all of the tests the windows in the attic remained open and those on the first story remained closed. For the purpose of cooling with outdoor air at night, the attic door and 11 windows on the second story were opened in the evening when the effective temperature outdoors became equal to the effective temperature indoors. The dampers were set and the basement door opened, so that the fan delivered outdoor air through the duct system until 7 a. m. The fan delivery was 2220 cfm or 9.4 air changes per hour. The windows and doors were closed at 7 a. m. and the dampers were changed so that the fan started delivering recirculated air and the outdoor air admitted for ventilation. The former was equivalent to 3.2 recirculations of the air in the house and the latter was equivalent to one air change per hour, making a total of 4.2 air changes per hour delivered by the fan. The fan was allowed to run continuously during the day and night.

In the first series of tests, which has been designated as Series 1-37, when the effective temperature of the indoor air on the second story rose to about 75 deg.

[2]Measuring Heat Transmission in Building Structures and a Heat Transmission Meter, by P. Nicholls (A.S.H.V.E. TRANSACTIONS, Vol. 30, 1924, pp. 65-104).

[3]Study of Summer Cooling in the Research Residence Using Water from the City Water Mains, by A. P. Kratz, M. K. Fahnestock, S. Konzo, and E. L. Broderick (A.S.H.V.E. TRANSACTIONS, Vol. 42, 1936, p. 193).

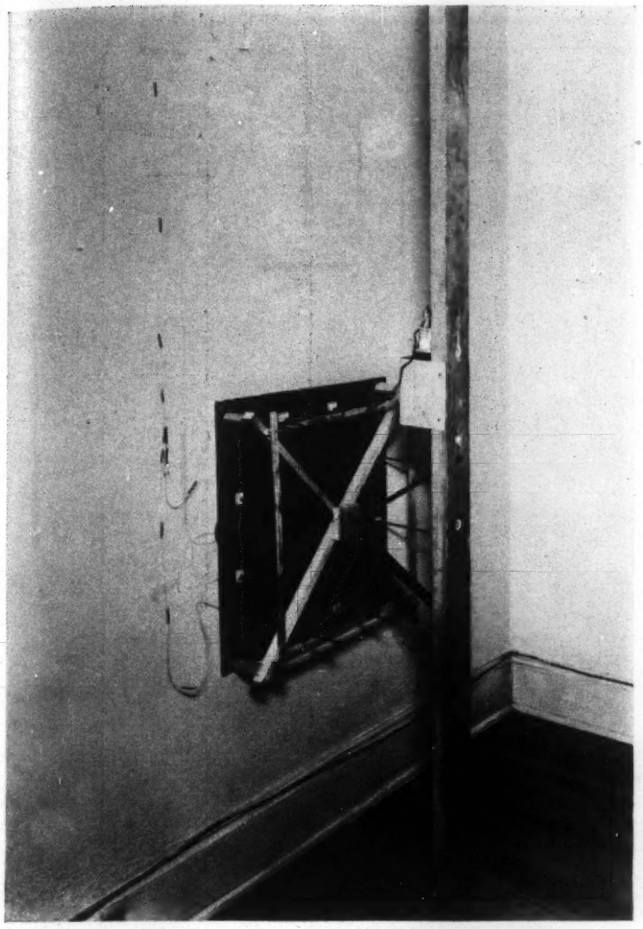

Fig. 2—Heat-flow meter installation on north wall

giving an average of approximately 74.5 deg for both stories, the condensing unit was started, and cooled air was distributed to the rooms on both stories. At the start of the test, when the relative humidity indoors was from 60 to 75 per cent, the corresponding dry-bulb temperature was from 79 to 78 F. The cooling plant was allowed to operate intermittently with the thermostat set to maintain approximately 78 F. However, the heat absorbing capacity of the unit was so small that when the rooms on both stories were being cooled the plant operated practically continuously until the time when cooling with outdoor air at night was started.

In the second series of tests, designated as Series 2-37, the recirculated air was distributed only to the first story rooms after 7 a. m., and the condensing unit was operated intermittently, maintaining an effective temperature of about 74 deg until 6:30 p. m. At that time the dampers were set so that all of the cooled air was distributed only to the second story rooms. Cooling with outdoor air at night was started when the effective temperature outdoors became equal to the effective temperature of the second story rooms.

Temperatures and heat flow measurements at the heat-flow meter installation were made at regular intervals, both day and night, for a period of 29 days. The data were plotted on a continuous chart and this chart was used as a basis for the analysis of results.

Results of Tests with Simultaneous Cooling on Both Stories

General Conditions

The operating characteristics of the cooling plant and a comparison of the actual and calculated cooling loads for the house can best be illustrated by the results obtained on a typical day. For this purpose the test made on September 1, 1937 was selected as typical of the operation obtained when the rooms on both stories were cooled (Series 1-37), and the results are shown in the first column of Table 1 and in Fig. 3. General results from all the tests are given in Table 2.

On the day which was selected as being typical for the tests made under conditions of operation for Series 1-37, the outdoor temperature was 89.3 F at 2 p. m. and reached a maximum of 91.0 F at 4 p. m. In this connection it is of interest to note that the maximum outdoor dry-bulb temperature for the entire season of 1937 was 95 F. The season was considerably cooler than the previous summer season, which was the hottest summer occurring in Urbana, Ill., since the year 1888, when the local Station of the U. S. Weather Bureau was established.

The air conditions at different locations are given in Items 5 to 15 of Table 1. The temperature of the cooled air leaving the registers was approximately 72.0 F. The rise in temperature of the air passing through the furnace casing and ducts was 3.3 F.

The cooling studies made during previous summers[4] had indicated that complete temperature traverses in the duct on both sides of the coil were required in order to obtain the proper corrections to be applied to the averages of the temperatures obtained from the wet- and dry-bulb thermometers at the two reading stations on each side of the coil. The distribution between the sensible and latent heat loads, as shown in Item 22, Table 1, was obtained by using the corrected values for the dry-bulb temperatures, and by multiplying the weighed amount of condensation from the coil by a constant representing the heat given up by the change in moisture content of the air per pound of water vapor condensed. The value of the constant amounted to 1050 Btu per pound of vapor condensed.

Operating Characteristics

The operating characteristics with cooled air supplied to the rooms on both stories are shown in Fig. 3. The second story windows had been open during the night previous to this test from 9:15 p. m. to 7 a. m., and outdoor air had been circulated by means of the basement fan. On this test the condensing unit operated intermittently from 9:07 a. m. to 11:30 a. m., and continuously from 11:30 a. m. to 10:07 p. m. maintaining the dry-bulb temperature between 77.0 F and 79.5 F.

At the start of the test at 9:07 a. m. the indoor relative humidity was 80 per cent and the indoor effective temperature was 75.5 deg. After the condensing unit had operated for about an hour the relative humidity decreased to 70 per cent with an accompanying effective temperature of 73.6 deg. However, during the off-period of the cooling unit both the relative humidity and ef-

[4]Loc cit. Note 3.

Table 1—Typical Operating Data and Results Obtained With Small-Sized Condensing Unit on August 17, and September 1, 1937

1. Date	9-1-37	8-17-37	8-17-37
2. Test No.	25	19	19
3. Series	1-37	2-37	2-37
4. Time of Day	2:00 p. m.	3:30 p. m.	7:30 p. m.
5. Outdoor Air			
Dry-Bulb, F	89.3	92.8	80.3
Wet-Bulb, F	74.3	75.8	71.3
Dew-Point, F	68.2	69.1	67.5
Relative Humidity, per cent	50.0	46.0	65.0
Specific Humidity, grains per pound dry air	103.8	107.0	101.2
6. Indoor Air, average breathing level temp.			
1st Story, F	78.4	78.1	79.4
2nd Story, F	78.1	83.0	81.0
7. Indoor Air, average at breathing level	Both Stories	1st St.	2nd St.
Dry-Bulb, F	78.2	78.1	81.0
Wet-Bulb, F	69.0	66.7	68.8
Dew-Point, F	64.6	60.7	63.0
Relative Humidity, per cent	62.9	55.0	54.0
Specific Humidity, grains per pound dry air	91.6	79.6	86.0
8. Indoor Air, entering return grille			
Dry-Bulb, F	78.0	76.6	79.2
Wet-Bulb, F	69.0	68.1	68.8
Dew-Point, F	64.8	63.9	63.8
Relative Humidity, per cent	64.0	64.8	59.0
Specific Humidity, grains per pound dry air	92.1	89.2	88.8
9. Ventilating Air			
Dry-Bulb, F	88.1	90.1	79.9
Wet-Bulb, F	75.1	74.1	72.7
Dew-Point, F	69.9	67.4	69.7
Relative Humidity, per cent	55.0	47.5	71.0
Specific Humidity, grains per pound dry air	110.4	101.0	109.6
10. Mixed Air, entering cooling coil			
Dry-Bulb, F	79.8	79.5	78.7
Wet-Bulb, F	70.2	69.7	69.4
Dew-Point, F	66.0	65.2	65.1
Relative Humidity, per cent	62.5	61.5	63.0
Specific Humidity, grains per pound dry air	95.8	93.4	93.2
11. Mixed Air, leaving cooling coil			
Dry-Bulb, F	68.7	67.2	67.3
Wet-Bulb, F	64.5	62.8	63.4
Dew-Point, F	62.3	60.3	61.2
Relative Humidity, per cent	82.0	78.5	81.0
Specific Humidity, grains per pound dry air	84.2	78.4	81.2
12. Air Temp. drop through cooling coil, F	11.1	12.3	11.4
13. Temp. cooled air leaving registers, average, F	72.0	70.5	72.0
14. Air Temp. rise in ducts and casing, F	3.3	3.3	4.7
15. Basement Air Temp. at breathing level, F	74.2	74.8	74.2

16. Quantity of air through wet coil	Both Stories	1st St.	2nd St.
Cubic feet per minute	978	916	958
Pounds per hour	4,400	4,120	4,310
Density of air, pounds per cubic feet	0.075	0.075	0.075
17. Number of air recirculations per hour	4.1	7.8	8.0
18. Ventilating Air			
Cubic feet per minute	231	228	229
Pounds per hour	1,040	1,026	1,032
Density of air, pounds per cubic feet	0.075	0.075	0.075
19. Cooling Coil			
Gross Face Area, square feet	2.07	2.07	2.07
Net Free Area, square feet	1.02	1.02	1.02
20. Air Velocity through Coil			
Gross Face, feet per minute	473	443	463
Free Area, feet per minute	959	898	940
21. Moisture condensed from air, pounds per hour	6.16	5.47	5.95
22. Heat given up by air; total Btu per hour	18,190	17,900	18,050
Heat due to moisture in air			
Btu per hour	6,470	5,740	6,250
Per cent of total heat	35.5	32.0	34.6
Sensible heat			
Btu per hour	11,720	12,160	11,800
Per cent of total heat	64.5	68.0	65.4
23. Water Temp. in condenser			
Inlet, F	59.7	59.8	59.5
Outlet, F	102.7	101.8	101.4
Rise, F	43.0	42.0	41.9
24. Water Temp. in house			
Inlet, F	59.5	59.6	59.5
Outlet, F	102.7	101.5	101.0
Rise, F	43.2	41.9	41.5
25. Quantity of cooling water			
Pounds per hour	448	442	451
Gallons per hour	53.7	53.1	54.1
26. Heat absorbed by cooling water, Btu per hour			
Through condenser	19,260	18,560	18,900
Through house	19,350	18,520	18,710
27. Refrigerant pressures			
Suction pounds per square inch	40	39	40
Discharge pounds per square inch	131	130	129
28. Nominal Rating of condensing unit with 55 F return refrigerant, 59 F condensing water and 74 F ambient air, Btu per hour	16,300	16,300	16,300
29. Compressor Speed, revolutions per minute	255	255	255
30. Compressor Motor Data			
Size, horsepower	2	2	2
Measured power rate, Kilowatt	1.37	1.34	1.34
31. Fan Speed, revolutions per minute	451	451	451
32. Fan Motor Data			
Size, horsepower	¾	¾	¾
Measured power rate, Kilowatt	435	440	440

fective temperature increased to values of 75 per cent and 75.3 deg respectively. This alternate decrease and increase of relative humidity and effective temperature was characteristic of the intermittent operation of the cooling plant. This characteristic was also observed in the case of the previous plants that had been installed in the Research Residence. A more detailed discussion of this condition is presented in a later section of this paper. The indoor comfort conditions during the intermittent periods of plant operation were on the upper border-line for comfort and could be regarded as reasonably satisfactory.

When the plant operated continuously, as from 11:30 a. m. to 10.07 p. m., the relative humidity stabilized at about 59 per cent, representing an effective temperature of 74 deg. The indoor conditions during these periods of continuous operation could be considered as entirely satisfactory. It is of interest to note that the minimum values of the indoor relative humidity that were maintained were approximately 10 to 15 per cent greater than those maintained during the tests made in the season of 1934 with a condensing unit having a heat absorbing capacity of practically 30,000 Btu per hour. To the extent that a relative humidity of 45 per cent might possibly be considered as slightly more desirable than a relative humidity of 59 per cent, even for the same effective temperature, the indoor conditions maintained with the larger condensing unit may be considered as slightly more favorable than those maintained with the smaller condensing unit.

It may be noted that in order to maintain the same effective temperature a unit having a smaller capacity should be operated to maintain an indoor dry-bulb temperature somewhat lower than that to be maintained with a unit having a larger capacity. This fact, together with the advisability of allowing for a rise in the indoor temperature on hot days when the cooling plant was insufficient in capacity to maintain a constant temperature, made it advisable to operate the smaller plant from a room thermostat which was set at approximately 78

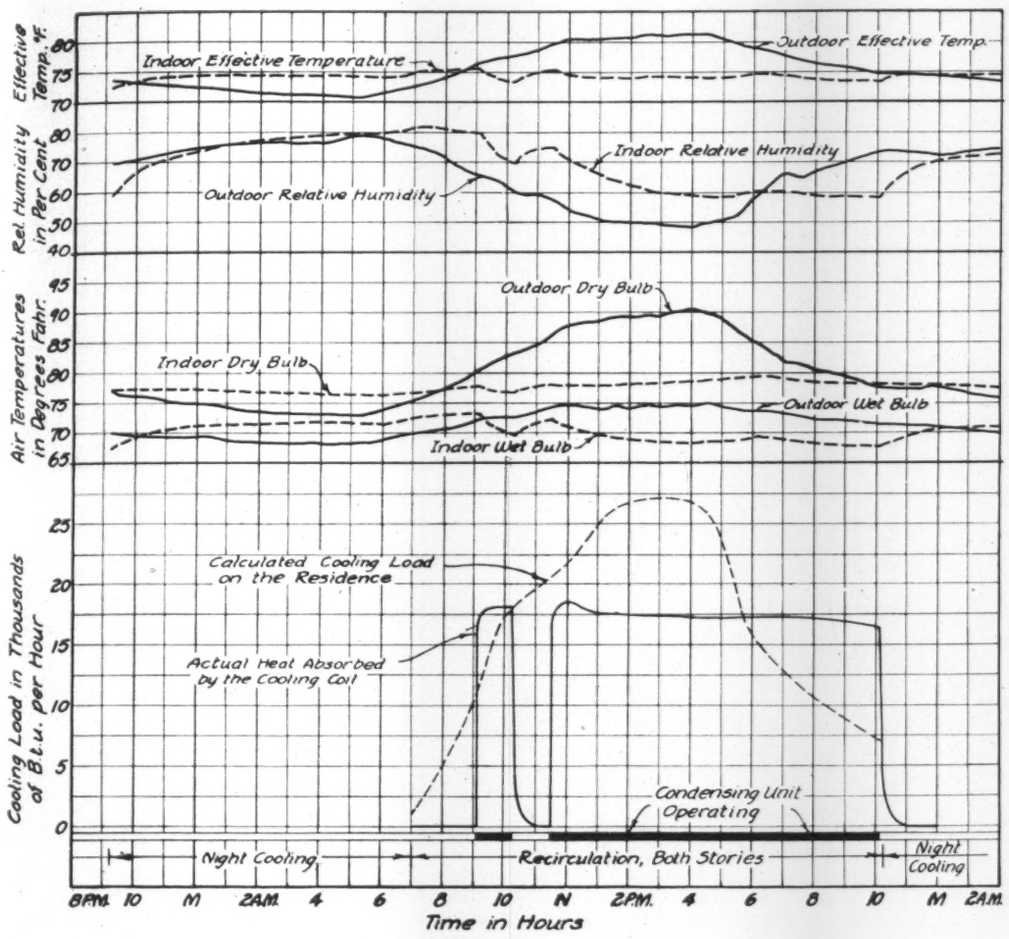
Effective Temp. °F.
Indoor Effective Temperature
Outdoor Effective Temp.
Rel. Humidity in Per Cent
Indoor Relative Humidity
Outdoor Relative Humidity
Air Temperatures in Degrees Fahr.
Outdoor Dry Bulb
Indoor Dry Bulb
Outdoor Wet Bulb
Indoor Wet Bulb
Cooling Load in Thousands of B.t.u. per Hour
Calculated Cooling Load on the Residence
Actual Heat Absorbed by the Cooling Coil
Condensing Unit Operating
Night Cooling
Recirculation, Both Stories
Night Cooling
8PM. 10 M 2AM. 4 6 8 10 N 2PM. 4 6 8 10 M 2AM.
Time in Hours

Fig. 3—Actual and calculated cooling load on Residence and air temperatures. Test No. 25, Series 1-37, September 1, 1937

F rather than 80 F. The differences in the relative humidities that were obtained have been attributed partly to the difference in seasons, partly to the difference in the mean coil temperatures and partly to the difference in the depth of the coils in the two plants. A complete discussion of the observed relationship between the mean coil temperatures and the relative humidity in the house has been presented in a previous paper.[5]

Limitations in Cooling Capacity

The cooling capacity of 17,300 Btu per hour was found to be sufficient for days in which the maximum outdoor temperature rose to 95 F and the minimum outdoor temperature during the night preceding the test did not exceed 71 F. Since the summer season was relatively cool, no tests were obtainable under severe weather conditions in which the outdoor temperature exceeded 95 F. From Fig. 3 it may be observed that during the period of continuous operation the indoor dry-bulb temperature gradually rose from a value of 78.1 F at 1 p. m. to a value of 79.5 F at 6 p. m. The indoor effective temperature at 6 p. m. was 74.8 deg which was just on the border-line for comfort. It is probable that if the outdoor air had been materially hotter the rise in the indoor temperature would have been greater and the indoor effective temperature would probably have been above that represented by the upper limit of the comfort zone. In the following section an analysis is presented of the limitations of the small condensing unit when used under all weather conditions and with the method of operation employed in the Research Residence.

In a previous study[6] of the effect of methods of night operation on conditions in the Residence on the following day, it was shown that the maximum indoor temperature that was attained during the day was dependent not only on the maximum outdoor temperature, but also on the minimum outdoor temperature that existed during the night previous to the test. For example, the maximum indoor air temperature of 81 F would result during the day, succeeding a night of operation with a given type of fan, if under upper limiting conditions a maximum outdoor temperature of 93.6 F was preceded by an outdoor minimum temperature of 62.0 F; and if under lower limiting conditions a maximum outdoor temperature of 83.6 F was preceded by an outdoor minimum temperature of 71.0 F. The same conditions could also be obtained with any maximum temperature lying between these limits when accompanied by a proper corresponding outdoor minimum temperature. It may be noted from this example that the average of the maximum and minimum outdoor temperature, or the daily median temperature, required in each case to result in the 81 F indoor air temperature was practically the same. This was also true of all of the daily median temperatures obtained from the intermediate combination of outdoor maximum and minimum temperatures required to result in this same indoor temperature. It is reasonable to expect, therefore, that the maximum indoor air temperature that is attained on a given day would bear some relation to the daily median temperature.

That such a relationship exists is shown by the data plotted in Fig. 4. These data were obtained during the summer of 1937 and show the maximum indoor tem-

[5]Study of Summer Cooling in the Research Residence Using Water at Temperatures of 52 F and 46 F, by A. P. Kratz, S. Konzo, and E. L. Broderick (A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, May and June 1937, pp. 319-327 and 391-400).

[6]Investigation of Summer Cooling in the Warm-Air Heating Research Residence, by A. P. Kratz, M. K. Fahnestock, and S. Konzo (*University of Illinois Engineering Experiment Station Bulletin*, No. 290, January 1937, pp. 73-80 and Fig. 25).

Table 2—Complete Data and Results for Tests

Date 1937	Test No.	Series No.	Weather Data: Degree-Hours Above 85 F	Degree-Hours Above 90 F	Total Sunshine in Hours and Minutes	Outdoor Air: Maximum Temp. During Test F	Minimum Temp. Previous to Test F	Average Temp. During Period Requiring Cooling: Temp. F	Hum. Per Cent	Indoor Air Conditions: At End of Night Cooling: Temp. F	Hum. Per Cent	Average During Period Requiring Cooling: Temp. F	Hum. Per Cent	Temperature Difference: Outdoor Max. to Average Indoor F	Average Outdoor to Avg. Indoor F
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
7–6	1	1a–37	19.4	0	15:00	89.0	65.0	84.3	47.9	73.7	71.0	79.5	54.0	9.5	4.8
7–7	2	1a–37	34.9	0.3	14:49	90.0	67.0	87.0	48.1	75.3	68.5	79.3	51.6	10.7	7.7
7–8	3	1a–37	19.0	0.1	7:15	90.0	75.0	86.4	50.2	76.4	69.6	78.7	58.9	11.3	7.7
7–9	4	1a–37	19.6	0	9:47	90.0	69.0	85.2	58.0	75.4	78.5	79.0	59.6	11.0	6.2
7–10	5	1a–37	45.1	4.0	13:07	91.0	70.0	87.9	56.8	77.1	79.4	79.4	58.0	11.6	8.5
7–11	6	1a–37	11.5	0	13:07	86.0	71.0	84.4	63.2	77.8	75.5	79.2	59.3	6.8	5.2
7–12	7	1a–37	2.1	0	5:32	84.0	70.0	79.0	72.0	77.8	74.3	78.9	66.3	5.1	0.1
7–14	8	1–37	23.6	0	14:28	88.0	68.0	86.0	66.9	76.4	87.3	79.6	65.0	8.4	6.4
7–15	9	1–37	22.6	0	10:55	88.0	73.0	86.1	62.6	78.3	78.8	79.8	61.2	8.2	2.7
7–16	10	1–37	25.6	0	11:18	89.0	72.0	86.2	58.4	78.0	78.1	79.7	59.9	9.3	6.5
7–24	11	1–37	16.6	0	7:58	88.0	67.0	85.9	57.8	75.0	70.5	78.6	59.7	9.4	7.3
8–4	12	1–37	0.0	0	1:35	83.0	68.0	80.7	72.9	75.4	74.6	79.0	72.1	4.0	1.7
8–6	13	1–37	1.0	0	11:48	85.0	68.0	82.5	58.8	73.8	68.3	78.5	60.0	6.5	4.0
8–7	14	1–37	33.9	0.7	14:07	91.0	65.0	87.5	56.4	73.8	74.1	79.1	65.3	11.9	8.4
8–9	15	1–37	0.6	0	6:46	88.0	68.0	83.3	60.7	76.1	81.8	78.8	61.4	9.2	4.5
8–10	16	1–37	13.3	0	10:10	90.0	66.0	84.3	60.9	74.4	75.5	79.1	60.6	10.9	5.2
8–11	17	1–37	35.6	0.6	9:40	91.0	70.0	87.8	58.0	76.5	77.0	79.7	62.0	11.3	8.1
8–16	18	1–37	35.5	1.2	13:47	94.0	65.0	85.9	55.6	73.1	71.9	79.2	60.5	14.8	6.7
8–17	19	2–37	47.1	12.8	11:34	95.0	72.0	86.7	55.4	77.2	77.9	78.6	59.6	16.4	8.1
8–18	20	2–37	19.4	0	8:42	91.0	68.0	86.7	59.4	75.3	63.3	78.5	66.6	12.5	8.2
8–19	21	2–37	27.0	0	13:40	92.0	71.0	86.5	50.6	76.1	73.5	78.8	59.2	13.2	7.7
8–20	22	2–37	21.0	0	12:41	90.0	67.0	84.7	59.3	75.4	68.5	78.7	61.2	11.3	6.0
8–30	23	2–37	12.6	0	9:51	91.0	68.0	83.2	59.4	75.6	73.0	78.5	61.0	12.5	4.7
8–31	24	1–37	6.7	0	6:15	87.0	69.0	83.7	60.4	76.2	76.1	77.7	63.3	9.3	6.0
9–1	25	1–36	27.1	0.3	8:45	92.0	72.0	85.2	58.6	77.0	81.6	78.1	66.5	13.9	7.1
9–2	26	1–37	23.6	0	8:35	91.0	72.0	85.7	53.9	77.2	78.8	78.8	60.5	12.2	6.9
9–3	27	1–37	12.6	0	9:57	89.0	70.0	84.2	58.6	76.4	80.2	78.1	62.0	10.9	6.1
9–4	28	1–37	0.5	0	5:15	87.0	72.0	81.3	65.6	77.0	78.3	77.6	65.2	9.4	3.7

peratures that were obtained when cooling was accomplished only by the use of outdoor air at night, and the condensing unit was not operated. The points representing these data are shown as circles in Fig. 4 and a full line is shown drawn through the points. It may be observed that for a daily median temperature of 75.5 F the maximum indoor temperature was 79.0 F.

The points representing the data obtained for the maximum indoor temperature when the small-sized condensing unit was operated are indicated by crosses, and a broken line drawn through the points represents the average. The intersection of the broken line with the full line is of particular significance as representing a critical daily median temperature, since it shows that for all values of the daily median temperature exceeding 75.5 F some amount of artificial cooling was required, and that for all values less than 75.5 F cooling with outdoor air at night alone was sufficient to maintain suitable indoor conditions. For example, for a daily median temperature of 83 F, if night air cooling alone was used, the indoor temperature would have risen to approximately 84 F; whereas if the small-sized condensing unit had been operated with the room thermostat set at 78 F, the indoor temperature would not have exceeded 80 F.

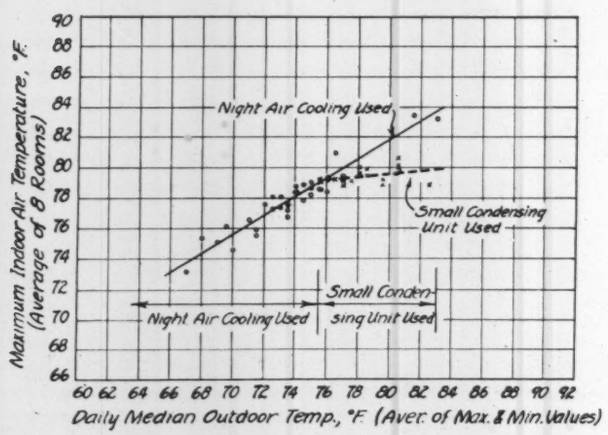

Fig. 4 — Relationship between indoor temperature and daily median outdoor temperature

The daily median temperature of 83 F, which was exceeded only once during the cooling season of 1937, may be considered as the probable upper limit for the range of application of the small-sized condensing unit. It is most probable that for daily median temperatures exceeding 83 F, the indoor temperatures would have risen above 80 F and the comfort conditions would not have been entirely satisfactory during the entire day. This is indicated by the fact that the broken line in Fig. 4 intersects the 80 F maximum indoor temperature line at a point corresponding to a daily median temperature of approximately 83 F. However, even with these higher outdoor temperatures, the resulting indoor conditions would have been much more acceptable than if no cooling had been used. Additional data under extreme weather conditions are desirable in order to investigate the actual resulting indoor comfort conditions.

From a statistical study of the daily median temperatures for a given season in Urbana, Ill., the number of days on which satisfactory indoor conditions would result, either from the use of air from outdoors at night or by supplementing such cooling with the use of the small condensing unit, can be determined by employing

Table 2—Complete Data and Results for Tests (Cont'd)

Operation of the Fan						Overall Test Period			Compressor Operation								
Night Air Cooling Previous to Test		Recirculation Previous to Test		Recirculation During Test Period		Start of Cooling with Condensing Unit to Start of Night Air Cooling			Time in Hours	Power Input Kwhr	Condenser Water			Total Moisture Condensed Lb	Heat Absorbed by Cooling Coil During Overall Test Period Btu			Ratio Moisture to Total Load
											Quantity in Gallons	Temperature						
Duration Hours	Power Kwhr	Duration Hours	Power Kwhr	Duration Hours	Power Kwhr	Start	End	Duration Hours				Inlet F	Rise F		Load Due to Moisture	Sensible	Total	
17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35
11.50	5.7	9.95	4.2	3.05	1.5	4:57 pm	8:00 pm	3.05	3.05	4.6	193	58.9	42.9	14.02	14,720	42,350	57,070	0.258
11.00	4.9	8.78	2.8	6.72	3.1	1:47 pm	8:30 pm	6.72	6.72	10.3	418	58.8	42.5	34.14	35,850	93,900	129,750	0.276
10.50	4.8	5.00	2.1	4.25	2.0	12:00 M	4:15 pm	4.25	3.52	5.7	223	58.9	42.3	24.66	25,890	46,750	72,640	0.356
14.75	6.8	5.50	2.2	7.50	3.5	12:30 pm	8:00 pm	7.50	6.28	10.1	416	59.0	42.1	47.31	49,680	76,150	125,830	0.395
11.00	5.0	3.28	1.3	12.22	5.4	10:17 am	10:30 pm	12.22	12.22	19.1	816	58.9	41.8	92.18	96,790	153,500	250,290	0.387
8.50	3.8	3.68	1.7	11.32	5.0	10:41 am	10:00 pm	11.32	11.32	17.9	757	58.9	41.3	84.70	88,940	140,900	229,840	0.387
9.00	4.1	2.50	1.0	7.50	3.3	9:30 am	5:00 pm	7.50	3.68	5.8	246	59.1	41.7	29.84	31,330	44,000	75,330	0.416
10.10	4.4	5.07	2.3	10.20	4.4	12:04 pm	10:16 pm	10.20	10.20	13.8	600	59.2	41.4	80.66	84,690	100,300	184,990	0.458
8.73	3.9	1.73	0.9	12.27	5.3	8:44 am	9:00 pm	12.27	12.27	16.3	713	59.4	41.0	87.61	91,990	126,800	218,790	0.421
10.00	4.6	2.63	1.1	12.12	5.4	9:38 am	9:45 pm	12.12	12.12	16.1	684	59.1	41.4	73.85	77,540	133,100	210,640	0.368
10.67	4.8	6.87	2.9	5.00	2.3	2:17 pm	7:17 pm	5.00	5.00	6.7	268	59.3	42.0	27.23	28,600	52,550	81,150	0.353
11.83	5.2	10.93	4.9	1.27	0.7	5:56 pm	7:12 pm	1.27	1.27	1.8	603	59.5	42.0	9.52	10,000	14,900	24,900	0.402
13.25	5.9	9.77	4.3	2.65	1.1	4:46 pm	7:25 pm	2.65	2.65	3.5	139	59.3	43.1	14.07	14,770	31,150	45,920	0.322
11.58	5.2	6.87	3.0	6.22	2.9	1:52 pm	8:05 pm	6.22	4.85	6.6	262	59.5	42.4	29.21	30,680	53,200	83,880	0.366
12.42	5.5	7.18	3.3	5.32	2.2	2:11 pm	7:30 pm	5.32	5.32	7.2	299	59.5	41.1	31.04	32,620	61,100	93,720	0.348
11.50	5.5	8.25	3.4	5.50	2.3	3:15 pm	8:45 pm	5.50	5.25	7.0	286	59.6	42.2	30.07	31,580	62,500	94,080	0.336
10.25	4.6	5.00	2.2	8.25	3.6	12:00 M	8:15 pm	8.25	8.25	11.3	459	59.4	42.7	52.44	55,100	91,300	146,400	0.377
12.17	5.5	8.50	3.7	4.50	2.0	3:30 pm	8:00 pm	4.50	4.50	5.9	244	59.5	42.3	25.72	27,000	52,900	79,900	0.338
11.00	5.1	3.08	1.2	11.92	5.3	10:05 am	10:00 pm	11.92	10.53	14.1	569	59.6	41.9	58.89	61,820	122,100	183,920	0.336
9.00	4.2	5.50	2.3	5.00	2.1	12:30 pm	5:30 pm	5.00	2.68	3.6	151	59.4	42.4	15.34	16,110	30,300	46,410	0.347
13.50	6.0	5.25	2.3	8.00	3.6	12:15 pm	8:15 pm	8.00	5.57	7.5	297	59.5	42.2	25.06	26,300	69,600	95,900	0.274
10.75	4.9	5.50	2.3	8.58	3.8	12:30 pm	9:05 pm	8.58	5.78	7.8	318	59.9	42.0	32.81	34,480	63,650	98,130	0.352
11.50	5.7	7.15	3.0	7.10	3.1	2:09 pm	9:15 pm	7.10	5.17	7.0	275	59.8	42.6	27.89	29,290	60,100	89,390	0.328
9.75	4.4	3.00	1.3	11.25	4.9	10:00 am	9:15 pm	11.25	11.00	15.0	586	59.4	43.2	69.29	72,750	123,000	195,750	0.372
9.75	4.5	2.12	0.8	13.13	5.7	9:07 am	10:15 pm	13.13	11.75	15.7	633	59.4	43.6	72.75	76,390	130,300	206,690	0.370
8.75	4.5	2.17	0.9	11.25	4.9	9:10 am	8:25 pm	11.25	11.08	14.9	592	59.4	43.1	62.45	65,600	128,400	194,000	0.338
10.58	4.9	3.00	1.2	12.00	5.2	10:00 am	10:00 pm	12.00	11.00	14.6	590	59.3	43.2	66.29	69,600	124,000	193,600	0.360
9.00	4.0	2.50	1.1	6.50	2.8	9:30 am	4:00 pm	6.50	6.00	8.0	324	58.9	43.3	40.90	42,950	61,400	104,350	0.412

the data shown in Fig. 4. Such a study has been applied to a season represented by the average of 11 seasons extending over the years of 1927 to 1937 inclusive and the results are shown in Table 3. This table shows a summary of the total number of days during the 11 year period for which the daily median temperatures assumed the values indicated. The data shown are for the total 1683 days included in all of the seasons between May 1 and September 30 (153 days per season) for the 11 years, and represent mild as well as severe summers. It may be noted that a daily median temperature of 75.5 F was obtained about 85 times during the 11 year period, averaging about eight days per season; whereas a daily median temperature of 83.0 F occurred 26 times during the 11 year period, or an average seasonal frequency of 2.4 days.

The data in Table 3, showing the average number of days during a season on which a given daily median temperature was attained, have been represented graphically in the upper part of Fig. 5. The broken line represents a mean of the points plotted. The values indicated on the right hand ordinate show the frequency in terms of percentage of the number of days in a season comprised of 153 days.

The lower curve in Fig. 5, representing the total number of days in the season having daily median temperatures not exceeding the value shown on the abscissa, was obtained by integrating the area under the curve included between the vertical ordinates of the origin and the stated value in the upper part of Fig. 5. It may be noted, for example, that for approximately 108 days in the total season of 153 days, the daily median

Table 3—Distribution of Days Corresponding to Indicated Values of the Median, or Average of the Outdoor Maximum and Minimum Temperature

(Data based on 1683 days included between May 1 and September 30 for years 1927 to 1937, inclusive.)[a]

Daily Median Temperature, F	40	41	42	43	44	45	46	47	48	49	50	51	52	53	54	55	56	57	58	59	60	61	62	63	64	65	66	67
No. of Days in 11-year Period	1	1	0	1	2	1	5	3	9	6	10	8	18	22	17	14	20	22	29	28	28	36	37	38	62	44	45	47
Avg. No. of Days per Season	0.09	0.09	0	0.09	0.18	0.09	0.45	0.27	0.82	0.55	0.91	0.73	1.64	2.00	1.55	1.27	1.82	2.00	2.64	2.55	2.55	3.27	3.36	3.45	5.64	4.00	4.09	4.27

Daily Median Temperature, F	68	69	70	71	72	73	74	75	76	77	78	79	80	81	82	83	84	85	86	87	88	89	90	91	92	93	94	95
No. of Days in 11-year Period	77	55	64	67	93	81	67	81	89	72	64	46	50	53	41	26	23	18	25	5	13	6	4	2	5	1	1	0
Avg. No. of Days per Season	7.00	5.00	5.82	6.09	8.45	7.36	6.09	7.36	8.09	6.55	5.82	4.18	4.55	4.82	3.73	2.37	2.09	1.64	2.27	0.45	1.18	0.55	0.36	0.18	0.45	0.09	0.09	0

[a]Data from Urbana, Ill. Station of United States Weather Bureau.

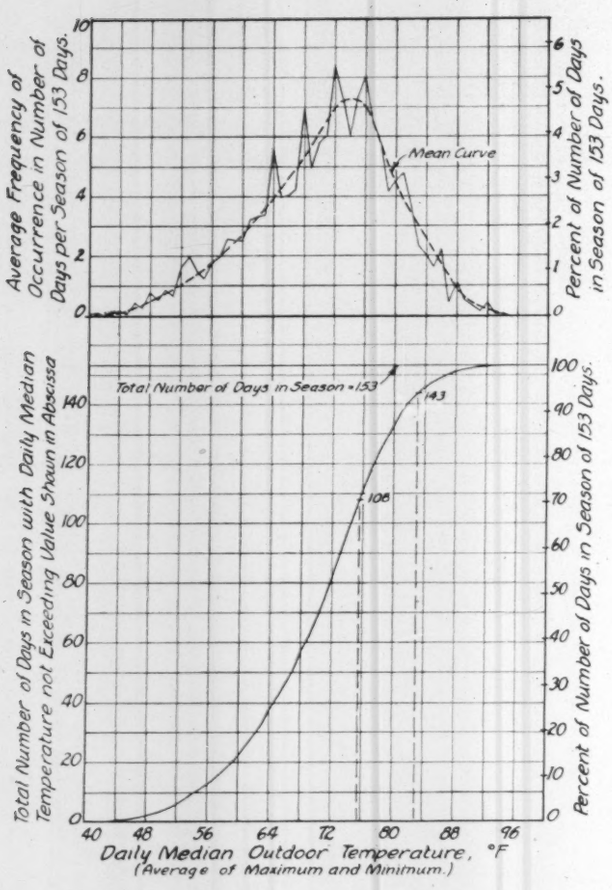
Average Frequency of Occurrence in Number of Days per Season of 153 Days.
Percent of Number of Days in Season of 153 Days.
Mean Curve
Total Number of Days in Season = 153
Total Number of Days in Season with Daily Median Temperature not Exceeding Value Shown in Abscissa
Percent of Number of Days in Season of 153 Days.
143
108
Daily Median Outdoor Temperature, °F
(Average of Maximum and Minimum.)

Fig. 5—Frequency of occurrence of days having stated daily median temperatures

temperature would be less than 75.5 F. The latter was indicated in Fig. 4 as representing a critical or limiting daily median temperature. Hence Fig. 5 indicates that cooling with air from outdoors at night alone would be sufficient to maintain comfortable conditions in the Research Residence for 108 days, or 70.6 per cent of the average season of 153 days; but would not be adequate for the remaining 45 days.

Similarly Fig. 5 indicates that for approximately 143 days in the season, the daily median temperature would not exceed the 83.0 F shown in Fig. 4 as the upper limit for the small condensing unit. That is, the small condensing unit, when used in connection with cooling with outdoor air at night by means of the basement fan, would be sufficient to maintain desirable comfort conditions in the Residence on 143 days, or 93.4 per cent of the season, but would not be sufficient in capacity to prevent the indoor temperature from exceeding 80 F on 10 days. These data represent a typical average season only, and considerable deviations from the average may occur during any given season. For example, during the relatively mild season of 1937 the daily median temperature exceeded 83.0 F on only one day; whereas during the extremely severe summer season of 1936, the daily median temperature exceeded 83.0 F on 28 days. Furthermore, the limitations shown in Fig. 4 apply only to the method of operation and to the operating conditions as maintained in the Research Residence. If a larger condensing unit had been used, the limiting value of the daily median temperature, as shown in Fig. 4, would have been greater than 83.0 F and, as indicated by Fig. 5, the unit would have had sufficient capacity to maintain acceptable conditions for more than 143 days per season.

The data presented indicate the feasibility of installing refrigerating equipment of small capacity in residences, particularly in those cases in which the initial cost of installation is a more important factor than the maintenance of ideal indoor conditions in the most severe weather. The data furthermore show the limitations resulting from the use of such a small condensing unit when operated under the conditions maintained in the Research Residence.

Acknowledgments

The results presented in this paper were obtained in connection with the summer cooling investigation (1937) in the Research Residence‡ at the University of Illinois conducted by the Engineering Experiment Station of which M. L. Enger, dean of the College of Engineering, is the director, and in the Department of Mechanical Engineering of which O. A. Leutwiler, professor of Mechanical Engineering Design, is the head. These results will ultimately comprise part of a bulletin of the Engineering Experiment Station. Acknowledgment is also due to R. B. Engdahl, research graduate assistant, and to R. B. Knight, special research graduate assistant, for services rendered in connection with the investigation, and to the various manufacturers who cooperated by furnishing instruments and equipment.

‡The Research Residence in Urbana, Illinois, was built, furnished, and completely equipped specifically for research work in warm air heating by the *National Warm Air Heating and Air Conditioning Association* in December, 1924.

(*Part 2 to be published in June*, 1938)

Air Conditioning Symbols

At a meeting held in New York on April 12, E. E. Ashley, A. S. H. V. E. representative on the committee, presented a report of the sub-group dealing with heating, ventilating, refrigeration and plumbing symbols to a General Committee on Mechanical Symbols of which T. E. French is chairman. After considerable discussion it was voted to submit the list of symbols to the Sectional Committee for their approval. Other members of the sub-group responsible for drafting the symbols were S. L. Land, L. L. Munier, J. L. Murphy, E. L. Penfrase, L. H. Roller, and E. R. Wolfert.

Definition of Air Conditioning

Stipulation No. 1883 of the Federal Trade Commission defines air conditioning in the following manner: The words *air conditioning* signify the simultaneous control by a mechanical device of various factors affecting both the physical and chemical conditions of the atmosphere within a given structure as a room, building, and the like, said factors including temperature, humidity and motion or circulation of the air within the structure. A device which does not control each and all of the designated factors is not properly represented, designated or referred to as an air conditioner within the purview and meaning of these words as generally understood by the trade and public.

Comparative Analysis of Office Building Air Conditioning Systems

By J. R. Hertzler* (*MEMBER*), York, Pa.

ACTUAL and potential field market studies for large commercial air conditioning equipment show office buildings to be of first importance. Each building warrants a careful analysis and study, if the proper system is to be installed to meet its requirements. A group of large, complete office building air conditioning systems have been analyzed in this paper for comparative purposes.

In order to show the wide variation in the types of equipment and methods of installation utilized, a group of 10 buildings, as indicated in Table 1, involving air conditioned areas of more than one million square feet have been used for analysis.

Table 1—Tabulation of Air Conditioned Buildings

Designation	Location	No. of Floors	Sq Ft Air Cond. Space
Building A	Houston, Tex.	8	169,712
Building B	Annex Bldg. to A, Houston, Tex.	15	80,960
Building C	Boston, Mass.	9	45,000
Building D	Boston, Mass.	11	97,700
Building E	St. Louis, Mo.	19	83,248
Building F	Fresno, Calif.	5	54,100
Building G	Cincinnati, Ohio.	15	60,910
Building H	St. Louis, Mo.	12	106,897
Building I	Hershey, Pa.	4	168,128
Building J	Wilmington, Del.	14	190,510

Factors Affecting Design

In the designing engineer's approach to office building air conditioning, several features are of primary importance to insure satisfactory acceptance, some of which are outlined herewith.

Type of Building

In order to properly calculate the heat gain it is important to know the shape of the building, whether the building is of windowless design or insulated, and whether of tower construction with exposures on all sides or partially shaded. It is also necessary to determine, within the enclosure, the partition arrangements, whether temporary or permanent, whether the spaces are rented or unrented and whether the hours of occupancy are uniform or variable throughout the building.

Zoning

This includes the separation of those spaces with similar exposures or similar heat gains so that uniform atmospheric conditions can be produced for all variations of load.

Apparatus Location

This factor is important from the standpoint of available space which can be sacrificed for the location of the air conditioning apparatus. This is closely allied with zoning and will have an important influence upon first cost as well as service and maintenance charges.

Air Distribution

The distribution of air is obtained in many different ways depending upon the architectural treatment and limitations. In practically all cases, proper air distribution will involve interior redecoration of existing buildings, thus increasing its importance to the owner.

Cooling Towers

Means of finally rejecting the heat removed from the conditioned spaces to the outside air is of extreme importance. Office buildings are located in metropolitan areas and consequently no cooling tower installation should render an inconvenience or nuisance to adjoining property. In selecting the location and type of cooling tower, the problems of (a) noise; (b) drift; (c) fogging; (d) appearance; (e) weight and structural supports; and (f) cost of operation are the principal considerations of the designing engineer.

Refrigeration

The medium furnished for dehumidification and cooling purposes must be of a type not to create hazards and should provide operating flexibility to economically and efficiently meet all conditions of load.

Heating

Combined with the cooling equipment should be means to provide, under automatic control, uniform and satisfactory conditions of temperature and humidity during the winter operation of the apparatus.

Features of Design

Always taken into account is the scope of automatic operation and control, types of air cleaning devices, possibility of need for independent humidity control in locations of excessive humidity or where moisture loads occur independent of dry or sensible heat gains, and other auxiliary items of peculiar importance in special cases.

Building A—Houston, Tex.

Type of Building. Existing office building is occupied by single tenant. Individual offices on each floor open to central corridors.

Zoning. Building is zoned east and west with individual floor control.

Apparatus Location. The 450 hp central water cool-

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ing plant is located in the basement. Central air washer for spaces up to and including 8th floor is located in the basement supplying dehumidified air to 2 local recirculating fans on each floor, one for east zone and one for west zone. Separate dehumidifier for 9th floor is installed on roof to supply the director's room and executive offices having roof exposure.

Air Distribution on lower floors utilizes corridor ceilings for location of duct work for supply and return air. In office space, ducts are arranged on face of columns to appear as false beams—with columns off center. Overhead plaque system of distribution is used on 9th floor with ducts in hanging ceiling. Director's room air supply is directed upward through radiator enclosures.

Cooling Tower. Because of the location of the building and the absence of nearby tall buildings, the cooling tower was located in the rear and on the first floor level.

Refrigeration. Two duplicate units of 225 hp size each are installed using dichlorodifluoromethane as the refrigerant. Each compressor is supplied with reduced capacity operating devices to permit power savings in direct proportion to reductions in load.

Heating of indirect type is furnished to supplement existing direct radiation and to furnish tempered, humidified, outside air during winter operation.

Auxiliary. Pneumatic temperature and humidity controls of conventional type, and throw-away type air filters are used in the design.

A compilation of technical data arranged by floors for this particular building is given in Table 2.

Building B—Houston, Tex.

Type of Building. An addition or annex 15 stories in height was added to the original building A and provisions for complete air conditioning were made in the plans and specifications as prepared. The occupancy was similar to that of Building A, but greater population density was made possible with the incorporation of air conditioning in the original design.

Zoning of the Annex Building B presented a difficult problem due to the unusual exposure. The south side of the building was shaded by Building A for the lower 9 floors, the number of separate zones having been selected as follows:

First Floor—2 zones—east and west.

Second Floor—3 zones—north, east and west.

Third-Tenth Floors—3 zones—north, east and west with a supplementary northeast zone added to the north or shaded zone.

Eleventh-Fourteenth Floors—4 zones—north, east, south and west.

Penthouse—4 zones—north, east, south and west.

Apparatus Location. Fan and air dehumidifier equipment of the spray type located in the basement supplies zones one, two and three of floors one to seven. Equipment on the roof supplies zones one, two, three and four of floors 8 to 15 inclusive. Separate supplemen-

Table 2—Technical Data by Floors, Building A, Houston, Texas

	1st Floor	2nd Floor	3rd Floor	4th Floor	5th Floor	6th Floor	7th Floor	8th Floor	9th Floor	Totals
Floor Area, Sq Ft		21,214	21,214	21,214	21,214	21,214	21,214	21,214	21,214	169,712
Ceiling Height, Ft		12.83	12.83	12.83	12.83	12.83	12.83	12.83	12.83	
Gross Room Volume, Cu Ft		272,200	272,200	272,200	272,200	272,200	272,200	272,200	254,500	2,159,900
Number of People		230	110	120	115	145	225	160	90	1,195
Floor Area per Person, Sq Ft		92.5	193	177	185	146	94	133	236	142.0
Watts of Light		5,050	3,800	4,280	5,750	4,350	4,600	5,100	4,400	37,330
Avg. Watts per Sq Ft Floor Area		0.24	0.18	0.20	0.27	0.20	0.22	0.24	0.21	0.22
Min. Outside Air, Cfm	Not	4,040	2,500	2,540	2,640	2,680	2,840	2,760	5,000	25,000
Min. Outside Air, Cfm per Person	Condi-	17.5	22.7	21.2	23.0	18.5	12.6	17.2	55.5	20.9
Min. Outside Air, Changes per Hour	tioned	0.89	0.55	0.56	0.58	0.59	0.62	0.61	1.18	0.70
Internal Sensible Tons		42.90	28.30	28.80	33.15	30.90	34.25	31.80	34.60	264.0
Internal Latent Tons		10.75	8.25	8.70	13.65	9.60	12.30	9.95	3.45	77.3
Total Internal Tons		53.65	36.55	37.50	46.80	40.50	46.55	41.75	38.05	341.3
Total Tons ÷ Sens. Tons = Int. Mult		1.25	1.29	1.30	1.41	1.31	1.36	1.31	1.10	1.29
Dew-point Required, deg F		56.8	56	55.8	53.25	55.6	54.5	55.6	59.0	56
Cfm from Dehumidifier		20,250	12,840	13,000	13,470	13,750	14,550	14,140	18,000	120,000
Circulated Air, Cfm		20,250	12,840	13,000	13,470	13,750	14,550	14,140	18,000	120,000
Cfm per Sq Ft Floor Area		0.95	0.60	0.61	0.63	0.65	0.68	0.67	0.85	0.71
Dry-Bulb Diffusion, Avg., deg F		23.2	24	24.2	26.7	24.4	25.5	24.4	21	24
Circulated Air Change, Min		13.4	21.2	21.0	20.2	19.8	18.7	19.3	14.1	17.9
Tons Refrigeration Required		71.37	48.44	49.47	59.08	53.02	59.61	54.37	59.32	454.68
Volume per Ton, Cu Ft		3,825	5,640	5,520	4,610	5,140	4,570	5,025	4,300	4,750
Floor Area per Ton, Sq Ft		298	449	431	360	400	356	392	359	374

Table 3—Technical Data by Floors, Building B, Houston, Texas

	1st Floor	2nd Floor	3rd Floor	4th Floor	5th Floor	6th Floor	7th Floor	8th Floor	9th Floor	10th Floor	11th Floor	12th Floor	13th Floor	14th Floor	1st Pent House	Totals
Floor Area, Sq Ft	7,850	6,160	5,620	6,500	6,240	7,040	6,630	6,280	5,440	6,400	3,920	3,800	3,760	3,300	2,020	80,960
Ceiling Height, Ft	16.0	12.83	12.83	12.83	12.83	12.83	12.83	12.83	12.83	12.83	12.83	12.83	13.83	15.83	14.83	
Gross Room Volume, Cu Ft	125,570	79,310	72,305	83,618	80,025	90,470	85,100	80,610	69,970	82,250	50,418	48,875	52,060	49,500	28,290	1,078,371
Number of People	46	70	136	86	89	89	89	89	89	66	49	52	179	106	31	1,266
Floor Area per Person, Sq Ft	170	88	41.4	75.5	70	79	74.5	70.5	61.0	97	80	73	21	31.1	65.1	64.0
Watts of Light	7,000	6,050	7,030	7,200	6,250	6,250	6,250	6,250	6,250	6,620	4,300	3,950	3,800	3,900	3,150	84,250
Avg. Watts per Sq Ft Floor Area	0.89	0.97	1.25	1.10	1.0	0.89	0.94	0.99	1.15	1.03	1.1	1.04	1.01	1.18	1.56	1.04
Min. Outside Air, Cfm	5,670	2,740	2,740	2,665	2,630	2,630	2,630	2,630	2,630	4,740	1,885	1,800	2,260	2,140	1,815	41,605
Min. Outside Air, Cfm per Person	123	39.2	20.2	31	29.6	29.6	29.6	29.6	29.6	71.6	38.4	34.6	12.6	20.2	58.5	32.9
Min. Outside Air, Changes per Hour	2.7	2.1	2.3	1.9	1.97	1.75	1.85	1.95	2.26	3.4	2.2	2.2	2.6	2.6	3.8	2.3
Internal Sensible Tons	23.60	11.40	11.40	11.10	10.93	10.93	10.93	10.93	10.93	19.70	7.85	7.50	9.44	8.92	7.58	173.14
Internal Latent Tons	0.59	0.89	1.73	1.10	1.13	1.13	1.13	1.13	1.13	0.84	0.62	0.66	2.28	1.35	0.39	16.10
Total Internal Tons	24.19	12.29	13.13	12.20	12.06	12.06	12.06	12.06	12.06	20.54	8.47	8.16	11.72	10.27	7.97	189.24
Total Tons ÷ Sens. Tons = Int. Mult	1.02	1.08	1.15	1.10	1.10	1.10	1.10	1.10	1.10	1.04	1.08	1.09	1.25	1.15	1.05	1.09
Dew-point Required, deg F	60	59.3	58.4	59	59	59	59	59	59	60	59.3	59.1	57	58.4	59.7	59.1
Cfm from Dehumidifier	13,600	6,350	6,100	6,100	6,030	6,030	6,030	6,030	6,030	11,400	4,380	4,150	4,350	4,780	4,300	95,660
Circulated Air, Cfm	17,514	7,607	6,863	6,531	6,861	6,790	7,161	6,628	7,362	11,516	4,942	4,885	6,352	4,867	4,300	110,179
Cfm per Sq Ft of Floor Area	2.24	1.23	1.22	1.00	1.10	0.96	1.08	1.05	1.35	1.80	1.26	1.28	1.69	1.47	2.13	1.36
Dry-Bulb Diffusion, Avg., deg F	15.5	17.3	19.2	19.7	18.4	18.6	17.6	19.2	17.1	19.8	18.4	17.6	15.7	21.3	20.3	18.0
Circulated Air Change, Min	7.15	10.4	10.5	12.8	11.7	13.3	11.9	12.2	9.5	7.1	10.2	10.0	8.2	10.1	6.6	9.7
Tons Refrigeration Required	47.7	24.6	25.3	24.2	23.9	23.9	23.9	23.9	23.9	40.1	17.4	16.9	21.1	20.1	16.6	373.5
Volume per Ton, Cu Ft	2,630	3,220	2,855	3,450	3,350	3,780	3,560	3,375	2,920	2,040	2,890	2,880	2,470	2,460	1,700	2,880
Floor Area per Ton, Sq Ft	164	250	222	268	260	294	278	262	228	268	225	225	178	164	122	216

Table 4—Technical Data by Floors, Building C, Boston, Mass.

	4TH FLOOR	5TH FLOOR	6TH FLOOR	7TH FLOOR	8TH FLOOR	9TH FLOOR	10TH FLOOR	11TH FLOOR	12TH FLOOR	TOTALS
Floor Area, Sq Ft	5,000	5,000	5,000	5,000	5,000	5,000	5,000	5,000	5,000	45,000
Ceiling Height, Ft	9.5	9.5	9.5	9.5	9.5	9.5	9.5	9.5	9.0	
Gross Room Volume, Cu Ft	47,500	47,500	47,500	47,500	47,500	47,500	47,500	47,500	45,000	425,000
Number of People	50	50	50	50	50	50	50	100	200	650
Floor Area per Person, Sq Ft	100	100	100	100	100	100	100	50	25	69.2
Watts of Light	5,000	5,000	5,000	5,000	5,000	5,000	5,000	5,000	5,000	45,000
Avg. Watts per Sq Ft of Floor Area	1	1	1	1	1	1	1	1	1	1.0
Min. Outside Air, Cfm	1,200	1,020	1,105	1,020	1,130	1,105	1,120	1,140	1,660	10,500
Min. Outside Air, Cfm per Person	24	20.4	22.1	20.4	22.6	22.1	22.4	11.4	8.3	16.2
Min. Outside Air, Changes per Hour	1.52	1.29	1.40	1.29	1.43	1.40	1.41	1.44	2.21	1.48
Internal Sensible Tons	6.08	4.22	4.22	4.22	4.72	5.21	7.10	7.91	15.75	59.4
Internal Latent Tons	0.64	0.64	0.64	0.64	0.64	0.64	0.64	1.27	2.55	8.3
Total Internal Tons	6.72	4.86	4.86	4.86	5.36	5.85	7.74	9.18	18.30	67.7
Total Tons ÷ Sens. Tons = Int. Multiplier	1.103	1.15	1.15	1.15	1.135	1.125	1.09	1.16	1.16	1.14
Dew-point Required, deg F	59	58.4	58.4	58.4	58.6	58.7	59.1	58.3	58.3	58.5
Cfm from Dehumidifier	3,350	2,820	3,045	2,820	3,105	3,050	3,090	3,140	4,580	30,500
Circulated Air, Cfm	4,390	3,685	4,000	3,685	4,075	3,990	4,050	4,125	6,000	38,000
Cfm per Sq Ft of Floor Area	0.88	0.74	0.80	0.74	0.82	0.80	0.81	0.83	1.12	0.85
Dry-Bulb Diffusion, Avg. deg F	15.6	12.9	11.9	12.9	13.0	14.7	19.7	21.6	29.6	16.9
Circulated Air Change, Min	10.8	12.9	11.9	12.9	11.6	11.9	11.7	11.5	7.50	11.2
Tons Refrigeration Required	9.50	7.22	7.42	7.22	7.98	8.41	10.34	11.82	22.14	92.1
Volume per Ton, Cu Ft	5,000	6,580	6,400	6,580	5,950	5,650	4,600	4,010	2,040	4,620
Floor Area per Ton, Sq Ft	526	693	675	693	626	595	484	423	226	488

tary volume control of the air supplied to each zone on every floor is furnished for accurate temperature regulation.

Air Distribution. Furred corridor ceilings are used for separate supply and return air ducts connected to the vertical risers in a ventilation stack provided in the new building.

Cooling Tower. In the tower of the new Annex Building B is located an enclosed concealed forced draft cooling tower of sufficient capacity to supply condensing water for the refrigeration equipment of both Buildings A and B.

Refrigeration. Two water cooling systems of the dichlorodifluoromethane type; one of 400 hp and one of 200 hp size, were furnished and installed in the basement to supply chilled water for dehumidification as required by the two air washers.

Heating. Indirect heating is supplied by the same system of duct work utilized for air distribution during the summer operating season.

Auxiliary. Pneumatic temperature and humidity control and throw-away type air filters are used in the installation.

A tabulation of the technical data for this particular installation is outlined in Table 3.

Building C—Boston, Mass.

Type of Building. Existing building of 13 stories is occupied by financial organizations in the Boston business district. It has 5000 sq ft per floor with internal as well as external offices.

Zoning. East and west zones per floor with individual volume control provide for load fluctuations on each of the relatively small floors.

Apparatus Location. The roof is utilized for the location of not only the spray type air dehumidifier and main supply fan, but for the evaporative condensers of the refrigeration plant as well. A 175 hp reciprocating compressor and shell and tube type water cooler are located in the basement.

Air Distribution. Air distributing duct work is located entirely within the building with supply and return risers.

Cooling Tower. Combined forced draft cooling tower and refrigerant condenser (evaporative type), is utilized to reject heat from the conditioned space. If city water at 80 F initial temperature were used, the cooling tower would normally require 1 per cent of 3 gpm per ton for make-up.

Refrigeration. One 175 hp water cooling unit is installed for the upper 9 floors, and an additional unit is to be furnished later for the lower 3 floors.

Heating. Direct radiation already installed was used, and supplemented with outside air tempering at the central supply fan.

Auxiliary. The automatic differential type pneumatic temperature and humidity control is readjusted to maintain a predetermined set of indoor temperatures for various outdoor conditions. An automatic by-pass control was added at the central air dehumidifier to compensate for changes in temperature of return air from the conditioned spaces. Throw-away type air filters were furnished for air cleansing.

In Table 4 are given condensed data applicable to this building.

Building D—Boston, Mass.

Type of Building. An existing office building offers space to prospective tenants and was only partially occupied at the time of the remodeling operation which included the installation of air conditioning.

Zoning. The north and the south wings are separately zoned. Individual field assembled surface cooling units are installed in each zone of the building, each unit supplying three floors.

Fig. 1—Location of supply outlet in Building D

Table 5—Technical Data by Floors, Building D, Boston, Mass.

	1st Floor	2nd Floor	3rd to 10th Floors Inc. Each Floor	11th Floor	Totals
Floor Area, Sq Ft	10,600	10,600	8,500	8,500	97,700
Ceiling Height, Ft	9.5	8.25	8.25	8.25	.....
Gross Room Volume, Cu Ft	100,000	87,500	70,000	70,000	817,500
Number of People	120	120	120	120	1,320
Floor Area per Person, Sq Ft	88.4	88.4	70.8	70.8	74.0
Watts of Light	7,000	3,500	3,500	3,500	42,000
Avg Watts per Sq Ft of Floor Area	0.66	0.33	0.41	0.41	0.43
Min. Outside Air, Cfm	2,200	2,200	2,200	2,200	24,200
Min. Outside Air, Cfm per Person	18.3	18.3	18.3	18.3	18.3
Min. Outside Air Changes per Hour	1.32	1.51	1.89	1.89	1.78
Internal Sensible Tons	7.81	8.29	8.29	16.0	98.4
Internal Latent Tons	1.53	1.53	1.53	1.53	16.8
Total Internal Tons	9.34	9.82	9.82	17.53	115.2
Total Tons ÷ Sens. Tons = Int. Mult	1.19	1.18	1.18	1.10	1.17
Dew-point Required, deg F	59.0	59	59	59.8	59.1
Cfm from Dehumidifier	7,000	7,000	7,000	12,300	82,300
Circulated Air, Cfm	7,000	7,000	7,000	12,300	82,300
Cfm per Sq Ft of Floor Area	0.66	0.66	0.82	1.45	0.84
Dry-Bulb Diffusion, Avg., deg F	12.5	13.2	13.2	14.5	13.4
Circulated Air Change, Min.	14.3	12.5	10.0	5.70	9.9
Tons Refrigeration Required	14.4	14.8	14.8	22.5	170.1
Volume per Ton, Cu Ft	6,950	5,900	4,730	3,110	4,800
Floor Area, per Ton, Sq Ft	736	715	574	378	574

Apparatus Location. Field assembled, surface cooling units located in the north wing of floors 2, 5, 8, and roof, supply the north zone of the following floors respectively. The second floor north unit supplies the north zone of floors 1 to 3. The fifth floor north unit supplies floors 4 to 5 north zone. The eighth floor north unit supplies the north zone of floors 7 to 9. The north unit on the roof supplies the north zone of the tenth and eleventh floors. Similar units located on the same floors supply identical floors in the south wing of the building.

Air Distribution. Duct work is installed within the office space rather than through the corridors and is furred to match the existing surroundings as indicated in Fig. 1.

The supply air is directed toward the outside walls of the room in each case.

Cooling Tower. A forced draft cooling tower is located on the roof to reject the heat removed from the building to the outside air by means of condensing water circulated through the refrigerant condensers in the basement. The cooling tower was located so that it was not visible from the street and surrounding buildings which are relatively lower in height; consequently its operation is not considered objectionable.

Refrigeration. A 250 hp water cooling system with two 125 hp compressor units and a single water cooler was located in the basement and functions to furnish the chilled water required for the surface cooling units located on the upper floors.

Heating. Outside air tempering is furnished through the heating coils in the field assembled air conditioning units located on the various floors, winter dehumidification being supplied through the use of atomizing sprays. The existing direct radiation in the building was allowed to remain and functions as previously installed.

Auxiliary. A differential system of automatic temperature and humidity control is furnished to automatically regulate the indoor temperature in accordance with the outside temperature utilizing automatic by-passing of recirculated air around the finned cooling surface of each air conditioner to maintain a uniform condition within the conditioned space during summer operation. Winter heating is also automatically controlled by a system of pneumatic control instruments.

Air filtering is furnished through the use of cleanable type air filters assembled in the field in conjunction with the fan and surface cooling and heating equipment.

Detailed figures are given in Table 5 for the unit factors applicable to Building D.

Building E—St. Louis, Mo.

Type of Building. At the time of installation of air conditioning in this 20-story building in St. Louis, a portion of the space was not rented, governing to some degree the selection of unitary surface cooling apparatus on each floor for providing air conditioning. Duct work is installed in all of the occupied zones, and on the unoccupied floors the air conditioning units have only been installed and connected to the refrigerating equipment. The installation on these floors is to be completed at such time as this space is rented, duct work then being located to suit the partition arrangement required by the tenant.

Zoning. On the lower floors, two air conditioning units are installed per floor, whereas, on the upper floors a single conditioner of adequate capacity is furnished. In general the zoning is east and west per floor with the individual units being automatically controlled for the independent control of temperature and humidity on each floor and in each zone.

Apparatus Location. Unit air conditioners of the factory assembled type, as illustrated by Fig. 2, are installed throughout the building, two units being furnished on the lower floors, and one unit on each of the upper floors. A central water cooling plant is located in the basement, with chilled water being supplied to the air conditioners for cooling purposes.

Air Distribution. Duct work is installed along the faces of the columns within the building and furred in without making an effort to balance the furred space on both sides of the column. The finished appearance is illustrated by Fig. 3.

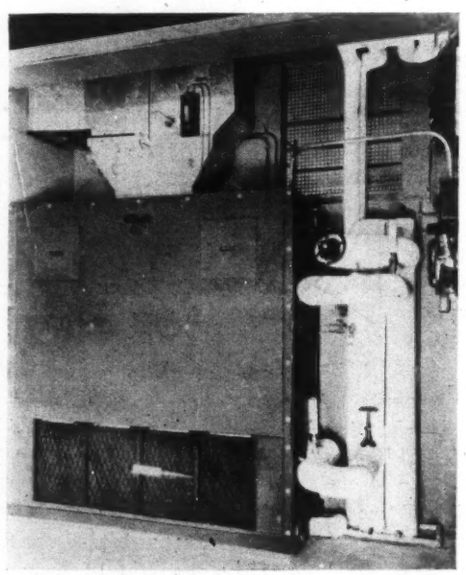


Fig. 2—Floor type conditioner in Building E

Fig. 3—Duct arrangement in Building E

Cooling Tower. A forced draft cooling tower is used for rejecting the heat removed from the conditioned space to the outside air.

Refrigeration. Refrigeration for dehumidification is furnished by a two compressor dichlorodifluoromethane water cooling system with one large water cooler operating in conjunction with a 200 hp and a 125 hp machine. This provides considerable flexibility in operation for the many and varied load factors involved in office building air conditioning.

Heating. The direct radiation in the building is supplemented with the tempering of outside air through the use of heating coils in the air conditioning units.

Auxiliary. A pneumatic system of temperature and humidity control is used in the building for year around temperature control.

Throw-away type air filters are furnished in the base of each vertical floor type air conditioner and are indicated clearly on Fig. 2.

A compilation of technical data arranged by floors for this particular building is given in Table 6.

Building F—Fresno, Calif.

Type of Building. In the exceedingly hot dry climate of Fresno, Calif., the use of evaporative cooling was originally deemed practical and such an installation was made by the owners. Later a mechanical refrigeration system designed along special lines was used to replace the original adiabatic cooling system.

There is a multiplicity of private offices arranged along the corridor of the property.

Zoning. The building is zoned principally into two zones per floor in the east and west wings, the building limitations deciding this factor to a large degree.

Apparatus Location. Surface coils for cooling the air supplied to the conditioned spaces are installed in the outlet of the existing adiabatic air washer. The refrigeration equipment is separated, with the evaporator coils of the direct expansion type being located on the inlet end of the main supply fan in the basement near the compressor, while the condenser is installed on the roof under the cooling tower.

Air Distribution. As the simplest system of air distribution was to be devised for this particular installation, Fig. 4 shows the method of applying building board below the existing furred ceiling, forming a plenum space used for distributing the conditioned air from the main supply fan. Openings were cut from the corridor at the ceiling line into each office above the entrance doors, with supply grilles being located as indicated on Fig. 5. The transoms above the entrance doors were returned to the factory in which they were made, so that their height could be reduced a sufficient amount to allow for the installation of the supply grilles as indicated. The building-board which forms the bottom of the air supply duct and the corridor ceiling is plastered on the corridor side and the upper side is lined with a $\frac{1}{16}$ in. asbestos sheet.

Cooling Tower. The cooling tower furnished for this installation serves the double purpose of furnishing water which is used to precool outside air prior to the cooling of this air by mechanical refrigeration. In addition the water leaving the outside air precooling coil is passed through the condenser of the refrigeration system. The cooling tower furnished is of the atmospheric type.

Refrigeration. Because such a large percentage of the heat to be removed from the building is in the form of sensible heat at a high temperature level, the use of cooling tower water for precooling reduces the refrigerating capacity necessitated for this installation to one-half that which would normally be required. It is also possible to make use of 100 per cent outside air and thus raise

Table 6—Technical Data by Floors, Building E, St. Louis, Mo.

	1st Floor	2nd Floor	3rd Floor	4th Floor	5th Floor	6th Floor	7th to 15th Floor Inc. Each Floor	16th Floor	17th Floor	18th Floor	19th Floor	20th Floor	Totals
Floor Area, Sq Ft	575		9,212	9,133	9,144	3,720	3,954	3,808	2,737	3,111	3,111	3,111	83,248
Ceiling Height, Ft	13.0		11.83	11.83	11.83	10.83	10.83	10.83	10.83	10.83	10.83	10.83	
Gross Room Volume, Cu Ft	7,475		109,000	108,000	107,000	40,400	42,850	42,400	29,620	34,750	34,750	34,750	933,795
Number of People	5		150	150	150	20	20	20	20	20	20	20	755
Floor Area per Person, Sq Ft	115		61.5	60.7	60.8	186	197	190	136	155	155	155	110.2
Watts of Light	800		22,000	22,000	22,000	2,000	2,000	2,000	2,000	2,000	2,000	2,000	96,800
Avg. Watts per Sq Ft Floor Area	1.39	Air	2.38	2.41	2.4	0.54	0.51	0.52	0.73	0.64	0.64	0.64	1.16
Min. Outside Air, Cfm	200	Condi-	2,000	2,000	1,700	800	800	700	800	800	800	800	17,800
Min. Outside Air, Cfm per Person	40	tioning	13.3	13.3	11.3	40	40	40	40	40	40	40	23.6
Min. Outside Air, Changes per Hour	1.61	Units	1.1	1.1	0.95	1.19	1.12	0.99	1.62	1.38	1.38	1.38	1.14
Internal Sensible Tons	1.75	In-	16.04	16.04	13.06	6.72	6.72	5.55	5.33	5.33	5.33	5.33	141.0
Internal Latent Tons	0.41	stalled.	2.88	2.88	1.79	1.20	1.20	0.94	0.94	0.94	0.94	0.94	24.7
Total Internal Tons	2.16	Duct	18.92	18.92	14.85	7.92	7.92	6.49	6.27	6.27	6.27	6.27	165.7
Total Tons + Sens. Tons = Int. Mult	1.23	Furn-	1.18	1.18	1.14	1.18	1.18	1.17	1.18	1.18	1.18	1.18	1.18
Dew-point Required, deg F	58.5	ished at	59	59	58.8	59.5	59.5	58.8	59	59	59	59	59.5
Cfm from Dehumidifier	1,200	Later	12,000	12,000	8,000	4,000	4,000	4,000	4,000	4,000	4,000	4,000	93,200
Circulated Air, Cfm	1,200	Date to	12,000	12,000	8,000	4,000	4,000	4,000	4,000	4,000	4,000	4,000	93,200
Cfm per Sq Ft of Floor Area	2.09	Suit	1.3	1.3	0.87	1.07	1.01	1.05	1.46	1.28	1.28	1.28	1.12
Dry-Bulb Diffusion, Avg., deg F	17.2	Tenants	15.7	15.7	19	19.4	19.4	16	17.7	17.7	17.7	17.7	13.0
Circulated Air Change, Min	6.22		9.1	9	13.4	10.1	10.7	10.6	7.4	8.7	8.7	8.7	10.0
Tons Refrigeration Required	2.7		24.4	24.4	19.5	10.1	10.1	8.4	9.25	9.25	9.25	9.25	217.4
Volume per Ton, Cu Ft	2,760		4,460	4,440	5,500	4,000	4,240	5,050	3,200	3,750	3,750	3,750	4,290
Floor Area per Ton, Sq Ft	213		378	374	468	368	391	454	296	336	336	336	383

Fig. 4—Plenum duct construction in Building F

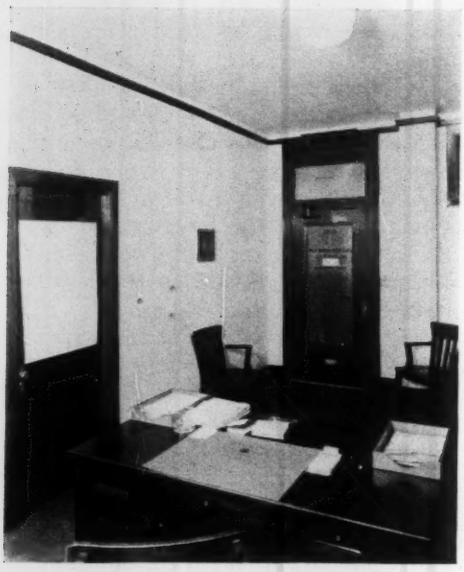


Fig. 5—Location of supply grille in Building F

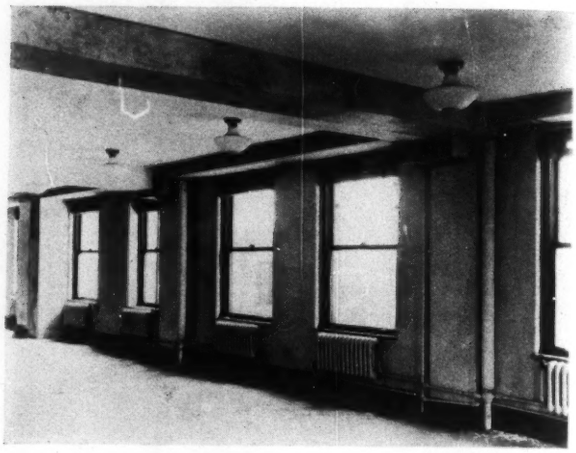


Fig. 6—Welded duct installed in Building G

Fig. 7—Duct work painted in Building G

the ventilation standard within the interior without a material penalty in operating cost. The 75 hp refrigeration machine with reduced capacity control is arranged to operate with direct expansion evaporator coils located in the basement.

Heating. The air conditioning equipment was furnished without provisions for operating during the winter.

Building G—Cincinnati, Ohio

Type of Building. A 15 story office building, consisting of 14 floors, was rented to individual tenants and one floor was used for office space by the owner. The lower 12 floors have an overall dimension of approximately 100 ft x 60 ft, the upper three floors form a tower, roughly 60 ft x 60 ft.

Zoning. Building is zoned east and west with separate recirculating fans on each floor receiving a supply of conditioned air from the central station dehumidifiers.

Apparatus Location. Two dehumidifiers of the spray type are installed, one being located on the fourth floor to take care of floors 1 to 5, the other being located on the 10th floor in an equipment room. This latter dehumidifier supplies air to recirculating fans on each of the upper 10 floors. A central water cooling system is located in the basement, chilled water being supplied to each of the two dehumidifiers under automatic control in accordance with heat gain requirements.

Local recirculating fans are used for the east and west zones, two fans per floor; they are located in the center stair wells adjacent to the elevator shaft.

Air Distribution. On this installation, an unusual type of system is utilized for the distribution of conditioned air over the space. The supply ducts were arranged along the outside building wall, with air being discharged towards the center of the building. The duct work is installed to appear as false beams, with square elbows being utilized throughout. The ductwork was reduced in size on the side of the outside wall, the space between the wall and the bottom of the duct being filled by a sheet metal separation.

Unique in the method of duct manufacture is the fact that the sheet metal joints were welded to appear as indicated in Fig. 6. The joints were then filled and two

Fig. 8—Room appearance before duct installation in Building G

coats of paint were applied directly to the sheet metal without the use of furring or a plaster finish. The finished installation is shown by Fig. 7 which can be compared with the original building appearance prior to the installation as indicated on Fig. 8.

City water is utilized for condensing purposes under automatic control of a gas pressure actuated water regulating valve, the quantity required being controlled in step with the automatic regulation of the refrigeration capacity.

Refrigeration. A 210 hp central station water cooling system was furnished with one 150 hp manually controlled compressor and one 60 hp automatic compressor. Each of the refrigerating machines is itself furnished with capacity reducing by-pass valves so that operation from 15 to 100 per cent capacity can be obtained with proportionate reduction in power input to the compressor motors for reduction in heat gain within the space.

Heating. Heating of the individual office space is obtained by the use of direct radiation. The air conditioning equipment is operated during the winter time to supply to the conditioned space tempered, humidified outside air for ventilation purposes.

Auxiliary. Differential thermostatic control of the pneumatic type is installed, with the summer indoor temperature automatically adjusted in accordance with outdoor temperature. Localized zone control is furnished by automatic damper regulation of the quantity of dehumidified air supplied to each of the local recirculating fans in the separate zones.

Throw-away type filters are furnished for supplementary cleaning of the air supply to the conditioned space.

In Table 7 are given complete technical data arranged by floors for this particular building.

Building H—St. Louis, Mo.

Type of Building. This 12-story building has an unusually large southern exposure with an exceedingly large percentage of the wall being glass. Sun effect or the heat gain due to the afternoon sun is the greatest single item of the total load and was, therefore, one of the governing factors of the design.

Zoning. Each floor of the building is designed with a separate recirculating fan. Supplementary control is furnished for four independent zones including a west sun zone, an east sun zone, a west shaded zone and an east shaded zone.

Apparatus Location. The layout of equipment utilized on this project required the use of no office floor space. Local recirculating fans for each floor are mounted from the ceiling.

The refrigeration equipment is located in the basement. In the space directly above, and in a court to the rear and outside, was built an air washer room with vertical supply and return duct risers located outside of the building.

Air Distribution. Duct work had to be run internally within the conditioned space and fitted around existing cross-beams and girders which, due to the shape of the building, interfered to a considerable degree. The finished duct work as furred-in with sound absorbing material is indicated in the photograph of the office space shown in Fig. 9. The location of the duct work in relation to the building columns is indicated in Fig. 9.

Cooling Tower. A cooling tower of the forced draft type is installed for cooling the condensing water utilized by the refrigeration plant.

Refrigeration. The dichlorodifluoromethane water cooling system furnished for this project is of 400 hp capacity and divided into two compressor units. Each 200 hp unit is operated in conjunction with a single

Table 7—Technical Data by Floors, Building G, Cincinnati, Ohio

	1st Floor	2nd Floor	3rd and 4th Floors Each Floor	5th Floor	6th to 8th Floors Inc. Each Floor	9th to 11th Floors Inc. Each Floor	12th Floor	13th Floor	14th Floor	15th Floor	Totals
Floor Area, Sq Ft	2,890	2,890	2,890	4,940	4,940	4,940	4,940	3,695	3,025	3,110	60,910
Ceiling Height, Ft	16.0	11.0	11.0	10.5	10.5	10.5	10.5	10.5	10.5	9.5	
Gross Room Volume, Cu Ft	46,300	32,800	32,800	51,500	51,500	51,500	51,500	38,800	31,800	35,240	662,540
Number of People	30	50	30	60	45	45	45	35	35	50	635
Floor Area per Person, Sq Ft	96.3	57.8	96.3	82.3	109.5	109.5	109.5	105.2	86.5	62.2	95.8
Watts of Light	5,150	6,000	6,000	8,800	8,800	8,800	8,800	5,650	5,400	5,400	110,000
Avg. Watts per Sq Ft Floor Area	1.78	2.08	2.08	1.78	1.78	1.78	1.78	1.53	1.78	1.73	1.81
Min. Outside Air, Cfm	1,085	770	770	1,205	1,205	1,205	1,205	906	743	825	15,509
Min. Outside Air, per Person, Cfm	36.16	15.4	25.6	20	26.8	26.8	26.8	25.9	21.2	16.5	24.4
Min. Outside Air, Changes per Hour	1.40	1.40	1.40	1.40	1.40	1.40	1.40	1.40	1.40	1.40	1.41
Internal Sensible Tons	6.47	5.02	4.63	7.47	6.68	6.98	7.82	5.60	4.92	7.11	94.0
Internal Latent Tons	1.15	1.19	0.93	1.62	1.44	1.44	1.44	1.10	0.98	1.02	19.0
Total Internal Tons	7.62	6.21	5.56	9.09	8.12	8.42	9.26	6.70	5.90	8.13	113.0
Total Tons ÷ Sens. Tons = Int. Mult	1.18	1.24	1.20	1.21	1.22	1.21	1.18	1.19	1.20	1.14	1.20
Dew-point Required, deg F	58	57	57.67	57.5	57.3	57.5	58	57.8	57.6	58.5	57.7
Cfm from Dehumidifier	3,250	2,415	2,285	3,670	3,400	3,580	4,100	2,910	2,540	3,825	46,600
Circulated Air, Cfm	4,320	3,240	2,960	4,890	4,326	4,530	5,380	3,820	3,200	5,184	62,522
Cfm per Sq Ft of Floor Area	1.50	1.12	1.02	0.99	0.88	0.92	1.09	1.03	1.06	1.66	1.03
Dry-Bulb Diffusion Avg., deg F	16.6	17.1	17.3	17.0	17.9	17.7	16.7	16.9	17.8	15.9	17.3
Circulated Air Change, Min	10.7	10.1	11.1	10.5	11.9	11.4	9.60	10.1	10.0	6.80	10.6
Tons Refrigeration Required	11.36	8.97	8.30	13.12	12.56	12.82	13.61	10.03	8.72	11.26	169.8
Volume per Ton, Cu Ft	4,070	3,660	2,960	3,920	4,100	4,000	3,780	3,860	3,650	3,140	3,900
Floor Area per Ton, Sq Ft	254	322	260	376	393	384	362	368	347	276	359

Fig. 9—Duct installation of Building H

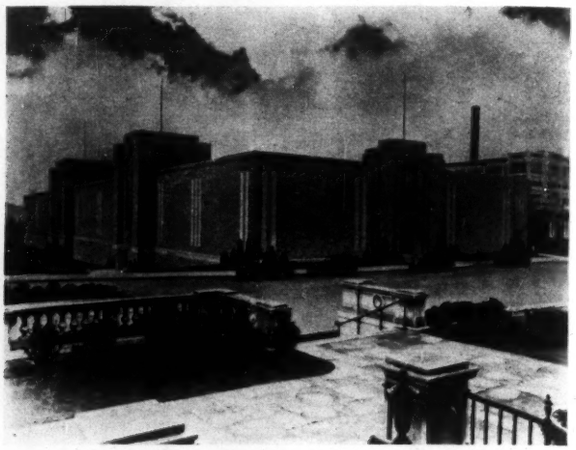


Fig. 10—Windowless office Building I

water cooler of the shell and tube type to provide chilled water for dehumidification during summer operation of the equipment.

Heating. Direct radiation was left in place and supplementary heating furnished by tempering outside air during winter operation.

Auxiliary. Automatic controls of the pneumatic type are provided. The recirculating fans located on each floor are furnished with supplementary automatic volume controls for the four zones on each floor to provide for further variation in heat gain.

Throw-away type filters are furnished for air cleaning to supplement the action of the spray type air dehumidifier.

Included in Table 8 are complete technical data arranged by floors for Building H.

Building I—Hershey, Pa.

Type of Building. A complete 4-story windowless office building, shown in Fig. 10, is used as the home office building of a large manufacturer. The walls are insulated and a water level is maintained on the roof to reduce the heat gain added by the sun's rays during summer operation.

Zoning. With insulated walls and the omission of windows, the needs for zoning are dependent almost entirely upon internal occupancy and internal loads. The basement and first floor are supplied by one air conditioning system, and the second and third floors by a second air conditioning system.

Apparatus Location. All of the mechanical equipment for refrigeration and air conditioning is located in the basement, with duct work being furnished to supply and recirculate the air to the several spaces.

Air Distribution. Various types of air distribution are combined in the several floors of the building; some duct work being exposed where its appearance is not objectionable, while some is installed to appear as false beams as illustrated in Fig. 11.

Well water is used for condensing purposes.

Refrigeration. Three 125 hp water cooling systems are installed in the basement to provide chilled water for the spray type dehumidifiers utilized in controlling the temperature and humidity.

Heating. Heating is accomplished entirely by an indirect method. The central station dehumidifiers are provided with tempering and reheat coils which provide the full amount of heat necessary in the conditioned space. This is made possible by the insulation of the walls and the fact that there are no windows or spaces with varying exposure. The wall insulation tends to reduce the amount of heat leakage to a minimum and the lack of windows practically eliminates localized infiltration of outside air.

Auxiliary. A differential system of pneumatic temperature and humidity control is provided to automatically adjust the indoor temperature in accordance with changes in outdoor temperature.

Throw-away type air filters are furnished for air cleaning.

Complete technical data for Building I and arranged by floors are given in Table 9.

Table 8—Technical Data by Floors, Building H, St. Louis, Mo.

	1st Floor	2nd, 4th, 8th, 10th and 11th Floors Each Floor	3rd, 5th, 6th and 7th Floors Each Floor	12th Floor	Floor 12-A	Totals
Floor Area, Sq Ft	8,401	8,938	8,081	8,413	4,075	106,897
Ceiling Height, Ft	16 75	10 0	10 0	10 0	9 0	
Gross Room Volume, Cu Ft	154,500	89,380	89,380	84,130	36,675	1,169,105
Number of People	120	120	120	120	75	1,515
Floor Area per Person, Sq Ft	70	74	67	70	54	70 5
Watts of Light	17,890	17,890	17,890	15,950	8,211	223 041
Avg. Watts per Sq Ft Floor Area	2 13	2 00	2 22	1 90	1 96	2 09
Min. Outside Air, Cfm	3,870	2,215	2,215	2,100	920	29,040
Min. Outside Air, Cfm per Person	32 2	18 6	18 4	17 5	12 3	19 2
Min. Outside Air, Changes per Hour	1 50	1 50	1 50	1 60	1 50	1 49
Internal Sensible Tons	26 9	17 4	17 4	21 6	19 7	242 8
Internal Latent Tons	2 20	2 00	2 00	2 00	1 30	25 5
Total Internal Tons	29 1	19 4	19 4	23 6	21 0	268 3
Total Tons ÷ Sens. Tons = Int. Mult	1 08	1 11	1 11	1 09	1 07	1 11
Dew-point Required, deg F	55 8	55 33	55 33	55 67	56	55 4
Cfm from Dehumidifier	13,650	9,100	9,100	11,150	9,900	108,500
Circulated Air, Cfm	15,200	13,100	13,100	15,800	15,000	177,400
Cfm per Sq Ft of Floor Area	1 81	1 56	1 62	1 88	3 68	1 66
Dry-Bulb Diffusion, Avg., deg F	20 2	15 0	15 0	15 6	15 0	15 5
Circulated Air Changes, Min	10 2	6 80	6 80	5 30	2 40	6 6
Tons Refrigeration Required	42 3	26 9	26 9	30 6	23 9	366 4
Volume per Ton, Cu Ft	3,660	3,320	3,320	2,750	1,540	3,180
Floor Area per Ton, Sq Ft	199	332	300	275	170	291

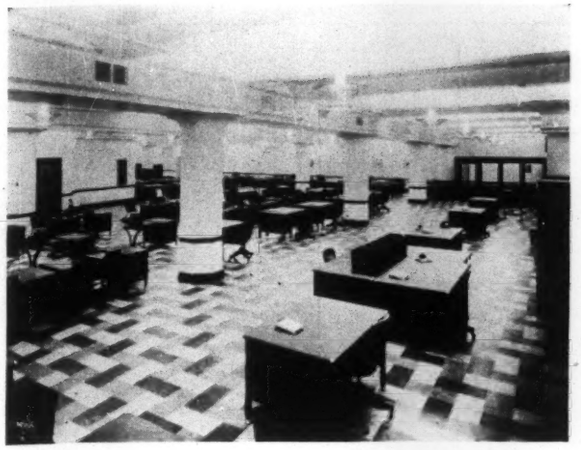


Fig. 11—Duct arrangement in Building I

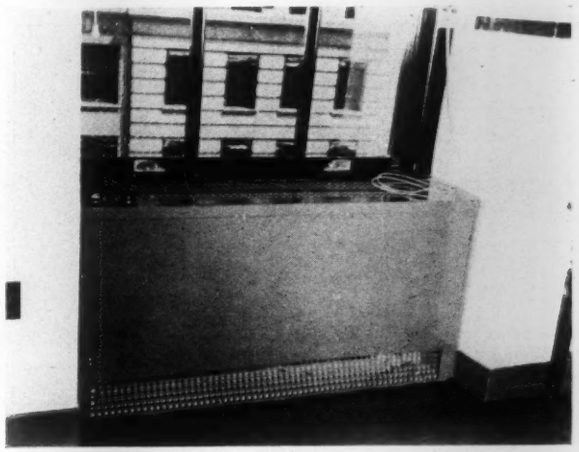


Fig. 12—Radiator enclosure in Building J

Building J—Wilmington, Del.

Type of Building. This particular building was newly constructed for occupancy by a single firm with the exception of the store space located in the lower floors which was to be leased to individual tenants.

In a new building of this type it was possible to install a system which did not interfere with the architectural treatment and that chosen in this particular case represents a unique design.

Zoning. Three separate zones were selected for the basement, first and second floors. One system was designed for the north stores, another system air conditioned south stores and a third system took care of the office spaces on the first and second floors.

The remainder of the building was zoned into north and south, with apparatus on the 14th floor taking care of the north portion of the third to thirteenth floors, while a separate unit on the 15th floor level is provided for the south half of the third to thirteenth floors.

Apparatus Location. An 800 hp water cooling system is located in the basement with 3 spray type air dehumidifiers to air condition the space up to and including the second floor. Dehumidifiers for floors three to thirteen are located on 14 to 15 floor levels, with the south zone being located on the fifteenth floor directly above the north zone unit on the fourteenth floor.

Air Distribution. On the lower floors an overhead plaque system of air distribution is installed in the conventional manner with duct work being concealed in the furred ceilings above the conditioned space.

On floors 3 to 13, radiator enclosures or air mixing units of the high pressure type are provided of such a design that considerable air from the room is mixed in the summer time with the cold supply air issuing from the orifices in the top of the radiator enclosure so that the temperature of the air stream 3 ft from the supply opening is at least 15 F higher than the temperature of air in the supply duct. A photograph of an actual radiator enclosure is illustrated in Fig. 12, which serves the double function of providing winter heating and also as a means of introducing a supply of conditioned air. The lattice grille opening at the floor line insures recirculation of room air at the floor level during winter operation. The rear openings nearest the window pane are directly above a finned direct heating surface located in the unit. The air is discharged vertically from these openings with room air being withdrawn from the floor line, passed over the heating surface and then out through the openings in the top of the unit to provide a current of warm air directly in front of the window pane.

The front row of openings on the room side of the enclosure and in the top of the unit, conceal oval-shaped air discharge orifices through which cold dehumidified air is discharged in the summertime and through which tempered, humidified, outside air is introduced in the wintertime. This front row of openings is the only supply of conditioned air to the room itself for temperature control and can be manually adjusted by a regulator provided in the top of the enclosure. The second control knob serves as a means of adjustment for the self-contained steam regulating valve for winter heating control.

Vertical duct risers from the third floor to the four-

Table 9—Technical Data by Floors, Building I, Hershey, Pa.

	Basement	1st Floor	2nd Floor	3rd Floor	Totals
Floor Area, Sq Ft	31,500	35,000	51,529	50,099	168,128
Ceiling Height, Ft	10.0	17.66	12.0	14.0	
Gross Room Volume, Cu Ft	315,000	618,120	621,350	701,400	2,492,106
Number of People	40	1,030	772	173	2,015
Floor Area per Person, Sq Ft	787	34	668	290	83.7
Watts of Light	94,500	94,158	115,536	102,750	406,944
Avg. Watts per Sq Ft Floor Area	3.0	2.69	2.24	2.04	2.42
Min. Outside Air, Cfm	3,065	5,315	6,485	3,380	18,245
Min. Outside Air, Cfm per Person	76.6	5.1	8.4	19.5	9.07
Min. Outside Air, Changes per Hour	0.58	0.52	0.62	0.28	0.44
Internal Sensible Tons	96.2	63.5	56.5	88.8	305.0
Internal Latent Tons	0.7	16.9	11.7	2.2	31.5
Total Internal Tons	96.9	80.4	68.2	91.0	336.5
Total Tons ÷ Sens. Tons = Int. Mult	1.00	1.26	1.21	1.02	1.10
Dew-point Required, deg F	61	61	52	53	60
Cfm from Dehumidifier	44,600	29,400	24,730	43,600	142,330
Circulated Air, Cfm	59,500	43,700	53,700	82,700	239,600
Cfm per Sq Ft of Floor Area	1.89	1.25	1.04	1.65	1.42
Dry-Bulb Diffusion, Avg., deg F	18	16	11.5	11.9	11.9
Circulated Air Change, Min	5.3	14.1	11.6	8.5	10.4
Tons Refrigeration Required	112.6	94.3	94.5	110.3	411.7
Volume per Ton, Cu Ft	2,800	6,550	6,590	6,350	6,050
Floor Area per Ton, Sq Ft	280	371	546	455	408
Inside Air Design Conditions	85 F DB 45% RH	85 F DB 50% RH	85 F DB 50% RH	70 F DB 50% RH	

Table 10—Technical Data by Floors, Building J, Wilmington, Del.

	Ground Floor	1st Floor	2nd Floor	3rd to 9th Floor Inc. Each Floor	10th Floor	Between 10th and 11th Floors Meeting Room	11th Floor	12th Floor	13th Floor	Total
Floor Area, Sq Ft	14,130	18,800	18,800	13,400	12,180	1,220	13,400	9,700	9,700	190,510
Ceiling Height, Ft	13.5	11.0	11.0	10.58	10.58	16.0	10.58	10.58	10.58	
Gross Room Volume, Cu Ft	190,600	206,800	206,800	141,800	128,500	19,520	141,800	102,200	102,200	2,071,520
Number of People	354	184	194	134	117	100	117	90	90	2,084
Floor Area per Person, Sq Ft	40	102	97	100	104	122	114	108	108	91.4
Watts Light	42,440	56,400	56,400	40,200	35,100	5,100	35,100	29,100	29,100	565,040
Avg. Watts per Sq Ft Floor Area	2.99	3.0	3.00	3.00	2.88	4.18	2.6	3.0	3.0	2.97
Min. Outside Air, Cfm	8,850	4,800	5,200	3,070	2,690	830	3,800	3,040	3,150	53,020
Min. Outside Air, Cfm per Person	25	26	26.8	22.9	23.0	8.3	32.4	33.8	35	25.4
Min. Outside Air, Changes per Hour	2.78	1.39	1.51	1.3	1.25	2.5	1.61	1.78	2.1	1.51
Internal Sensible Tons	36.2	25.5	27.9	18.5	16.2	5.0	22.9	18.3	18.9	295.4
Internal Latent Tons	4.6	3.9	4.3	1.7	1.4	1.3	1.4	1.1	1.1	29.7
Total Internal Tons	40.8	29.4	32.2	20.2	17.6	6.3	24.3	19.4	20.0	325.1
Total Tons ÷ Sens. Tons = Int. Mult	1.13	1.18	1.18	1.09	1.09	1.26	1.06	1.06	1.06	1.10
Dew-point Required, deg F	58.5	58.0	58	58	59	56.5	57.6	57.6	57.6	59.0
Cfm from Dehumidifier	18,650	13,500	14,700	9,210	8,050	2,500	11,400	9,100	9,480	149,350
Circulated Air, Cfm	18,650	13,500	14,700	9,210	8,050	2,500	11,400	9,100	9,480	149,350
Cfm per Sq Ft of Floor Area	1.32	0.72	0.78	0.69	0.66	2.0	0.85	0.94	0.97	0.78
Dry-Bulb Diffusion, Avg., deg F	21.5	22	22	22	21	23.5	22.4	22.4	22.4	21
Circulated Air Change, Min	10.2	15.3	14.1	15.4	16.0	7.8	12.4	11.2	10.8	13.9
Tons Refrigeration Required	65.5	48.0	52.0	31.4	25.5	7.9	39.9	31.8	33.1	515.6
Volume per Ton, Cu Ft	2,910	4,300	3,980	4,500	5,050	2,480	3,550	3,220	3,100	4,020
Floor Area per Ton, Sq Ft	216	392	362	426	476	155	336	305	293	369

teenth floor are run along the outside walls adjacent to the air mixing units located under the windows and connections are made from the vertical duct risers to each unit. The duct risers are arranged along the outside wall adjacent to columns.

Cooling Tower. A forced draft cooling tower is furnished for reclaiming the condensing water utilized by the refrigeration system installed for summer dehumidification purposes.

Refrigeration. A water cooling system of 800 hp size with two 400 hp compressors is installed in the basement. Chilled water is circulated from the refrigerating system through heat exchanging surfaces in the spray type air dehumidifiers. Spare refrigeration capacity is provided for future extension of the air conditioned space.

Heating. Winter heating is obtained by the use of finned heaters concealed within the radiator enclosures which serve as a means for air distribution summer and winter. Dehumidified air tempered to the room temperature is supplied to the air mixing units through the duct work used in the summertime for cooling.

Auxiliary. A differential type of temperature humidity control is utilized in which the indoor temperature is automatically regulated in accordance with the outside street temperature.

Automatic, oil type filters are furnished for air cleaning.

In Table 10 are given complete technical data for Building J arranged by floors.

Composite Summary of Technical Data

In further explanation of the technical data submitted for each of the foregoing buildings, certain of the items tabulated may be further amplified.

The item *floor area in square feet* includes only the net air conditioned space to which air is supplied from the system.

The *minimum outside air cfm* is that quantity of outside air introduced to the air conditioning system under a condition of maximum load at the design point tabulated in each case. This is the quantity of outside air including infiltration which has been estimated to be introduced to the building to maintain a definite standard of air purity within the enclosure.

The *internal sensible tons* is the internal sensible heat gain within the conditioned space including heat transmission through the walls and glass, heat from electric lighting, sensible heat from people and any other source of dry heat within the enclosure itself, which is a component part of the internal sensible heat gain. The internal sensible load also includes that small portion of the outside air which infiltrates direct to the conditioned space thereby adding to the internal sensible heat load.

The *internal latent tons* includes the internal latent heat load within the conditioned space including the latent heat of infiltrated outside air exclusive of outside air introduced into the air conditioning apparatus for ventilation purposes.

The term *internal multiplier* which is defined as *total tons divided by sensible tons* is the ratio of total internal heat to internal sensible heat. This relationship is set up to determine the dew-point of the air required to be supplied to the conditioned space for maintaining the predetermined condition of temperature and humidity for the calculated internal, sensible and latent, heat load.

The term *dry bulb diffusion average* is the temperature rise experienced by the supply air from the time it leaves the supply register until it is withdrawn from the room through the return air grille. This term *diffusion* is also the temperature range through which a specified quantity of circulated air would be warmed in absorbing the internal sensible heat load in the conditioned space.

The *circulated air change in minutes* is the number of minutes required for the complete room volume to be displaced by the supply fan air quantity. This is, therefore, the cubical content of the building divided by the fan air capacity in cubic feet per minute.

The term *tons refrigeration required* is the internal total sensible and latent load plus the load due to outside air introduced under maximum conditions for ventilation purposes.

The *cubic feet volume per ton* and *square feet floor area per ton* are based upon the particular values as tabulated insofar as they relate to the total refrigeration required.

A study of the technical data in Tables 2 to 10 inclu-

Table 11—Composite Summary of Technical Data

	BUILDING A	BUILDING B	BUILDING C	BUILDING D	BUILDING E	BUILDING F	BUILDING G	BUILDING H	BUILDING I	BUILDING J
Total Number of Floors	10	18	13	12	20	7	17	13	4	15
Number of Floors Conditioned	8	15	9	11	19	5	15	12	4	14
Outside Air Design Conditions	100 F DB-79 F WB	100 F DB-79 F WB	95 F DB-75 F WB	95 F DB-75 F WB	100 F DB-76 F WB	100 F DB-70 F WB	95 F DB-78 F WB	95 F DB-78 F WB	95 F DB-76 F WB	95 F DB-78 F WB
Inside Air Design Conditions	80 F DB-50% RH	80 F DB-50% RH	80 F DB-50% RH	80 F DB-50% RH	80 F DB-50% RH	80 F DB-45% RH	80 F DB-50% RH	80 F DB-45% RH	a	80 F DB-50% RH
Floor Area, Sq Ft	159,712	80,960	45,000	97,700	83,248	54,100	60,910	106,897	168,128	190,510
Room Volume, Cu Ft	2,159,900	1,078,371	425,000	817,500	933,795	616,740	662,540	1,169,105	2,492,106	2,071,500
Number of People	1,195	1,266	650	1,320	755	400	635	1,515	2,015	2,084
Floor Area per Person, Sq Ft	142.0	64.0	69.2	74.0	110.2	135.2	95.8	70.5	83.7	91.3
Watts of Light	37,330	84,250	45,000	42,000	96,800	6,000	110,000	223,041	406,944	565,040
Avg. Watts per Sq Ft of Floor Area	0.22	1.04	1.0	0.43	1.16	0.11	1.81	2.09	2.42	2.97
Min. Outside Air, Cfm	25,000	41,605	10,500	24,200	17,800	40,000	15,509	29,040	18,245	53,020
Min. Outside Air, Cfm per Person	20.9	32.9	16.2	18.3	23.6	100	24.4	19.2	9.07	25.4
Min. Outside Air, Changes per Hour	0.70	2.32	1.48	1.78	1.14	3.88	1.41	1.49	0.44	1.54
Internal Sensible Tons	264.0	173.1	59.4	98.4	141.0	72.2	94.0	242.8	305.0	295.4
Internal Latent Tons	77.3	16.1	8.3	16.8	24.7	5.1	19.0	25.5	31.5	29.7
Total Internal Tons	341.3	189.2	67.7	115.2	165.7	77.3	113.0	268.3	336.5	325.1
Total Tons÷Sens. Tons=Avg. Int. Mult.	1.29	1.09	1.14	1.17	1.18	1.07	1.20	1.11	1.10	1.10
Circulated Air, Cfm	120,000	110,179	38,000	82,300	93,200	40,000	62,522	177,400	239,600	149,350
Cfm per Sq Ft of Floor Area, Avg.	0.71	1.36	0.85	0.84	1.12	0.74	1.03	1.66	1.42	0.78
Circulated Air Change, Minutes Avg.	17.9	9.2	11.2	9.9	10.0	15.4	10.6	6.6	10.4	13.8
Tons Refrigeration Required	454.68	373.5	92.1	170.1	217.4	144.3	169.8	366.4	411.7	515.6
Volume per Ton, Cu Ft	4,750	2,880	4,620	4,800	4,290	4,260	3,900	3,180	6,050	4,020
Floor Area per Ton, Sq Ft	374	217	488	574	383	375	359	291	408	369
Equipment Summary										
Refrigeration—Compressors	2-14″x10″ 4 Cyl.	15″x13½″ 4 Cyl. 15″x13″ 2 Cyl.	6103 FE	2-12½″x10″ 2 Cyl.	7108-F 798-F	6751-F	14″x10″ 2 Cyl. 9″x7½″ 2 Cyl.	2-7108-F	2-14″x10″ 2 Cyl. 1-15″x10″ 2 Cyl.	2-610F2-F
Condensers	2-43½″x16′	4365 and 6065	3 EFB 2000	2 12″x15′	31,251	1,202	43″x16′	41,251	3-38″x16′	81,251
Coolers or Direct Exp.	2-43¾″x16′	3364 4864		38″x16′	3,806	Direct Exp.	38″x16′	4,303	3-28″x16′	2-4306
Cooling Tower	Forced Draft	Forced Draft	None	Forced Draft	Forced Draft	Atmospheric	None	Forced Draft	None	Forced Draft
Air Conditioning Equipment	2 Base Washers Supply Floors 2-8 Penth. Washer Supplies 9th Floor	Base Washer Supplies Floor 1-7 Penth. Washer Supplies 8-1st. P. H.	Washer on Roof Supplies Floors 4-12	2 Units on 2nd, (1st-3rd.) 2 Units on 5th, (4th-6th.) 2 Units on 8th, (7th-9th.) 2 Units on Roof, (10th and 11th)	Air Cond. Units on Each Floor	F-12 and Water Coils in Basement Supply Floors 2-6	4th Floor Washer Supplies Floors 1-5 10th Floor Washer Supplies 6-15 Floors	C. S. Air Washer 1st Floor	Washers in Base Supply Base to 3rd Floors	3 Base Washers Supply Base. 2nd Floor Washers on 14th and 15th Serve Floors 3-13
Zones	2 Recirc. Fans Each Floor 2-8 Separate Supply to 9th Floor	3 Recirc. Fans For Floors 1-10 4 Recirc. Fans for Floors 11-1st. P. H.	By-pass Control at Washer Supplementary Vol. Control Each Floor	By-pass Control at Each Unit Supplementary Vol. Control Each Floor	Air Cond. Units Each Floor Surface Type	2 Zones on Each Floor Volume Control	2 Recirc. Fans Each Floor	1 Recirc. Fan per Floor Supply Volume Control 4 Zones Each Floor	1 Washer Base. and 1st Floor 1 Washer 2nd, 3rd Floor CS By-pass Control	Air Mixing Units in Rooms By-pass Control and Supply Manual Control
Heating	O. S. Air Temp. Indirect	O. S. Air Temp. Indirect	Direct Rad. O. S. Air Temp.	Direct Rad. O. S. Air Temp.	Direct Rad. O. S. Air Temp.	—	Direct Rad. O. S. Air Temp.	Direct Rad. O. S. Air Temp.	O. S. Air Temp. Indirect	Direct O. S. Air Temp. Indirect
Automatic Controls	Pneu.	Pneu.	Diff.-Pneu.	Diff.-Elec.	Pneu.	Pneu.	Diff.-Pneu.	Pneu.	Diff.-Pneu.	Diff.-Pneu.
Filters	Throw Away	Throw Away	Throw Away	Cleanable	Throw Away		Throw Away	Throw Away	Throw Away	Auto. Oil
Miscellaneous						C. T. Water Used For O. S. Air Precooling				

[a]See under technical data by floors.

sive will show the floor to floor variation of typical office buildings. These values are totaled or averaged and compiled in the composite summary of Table 11 comparing the 10 air conditioning systems on an overall basis.

This composite summary reveals that insofar as population density is concerned this factor varies over a wide range from a minimum of 64 sq ft of floor area per person in Building B to 142 sq ft of floor area per person in Building A.

The average lighting in watts per square foot of floor area also varies over a wide range and is usually the amount of lighting required simultaneously with maximum sun effect on a given building. This particular value has a tendency to be increased due to possibility of widespread adoption of indirect lighting systems of extreme intensity which will have a definite bearing upon the required capacity of the air conditioning plant. The maximum average lighting load simultaneous with maximum sun effect is that tabulated for Building J or 2.97 w per square foot of floor area.

As for outside air supply furnished for ventilation purposes, the normal minimum infiltration will generally supply in a typical office building a liberal amount of outside air for maintaining an unusually high ventilation standard. Where the quantity of outside air is definitely controllable, as is also the population density within the space, the quantity of outside air introduced in cubic feet per minute per person can be more closely regulated. The windowless office Building I in which infiltration can be reduced to an absolute minimum through the elimination of windows, was installed using 9.07 cfm of outside air per person against the maximum outside air quantity computed for Building F of 100 cfm per person.

The minimum outside air supply ranges from 0.44 changes per hour in the windowless Building I to 3.88 changes per hour in Building F where climatic conditions economically permitted the use of 100 per cent outside air.

The air circulated through the conditioned space ranged from 0.71 cfm per square foot in Building A to 1.66 cfm per square foot in Building H, the latter quantity being required by the unusually large portion of glass in the walls and the large exposure to sun effect.

The circulated air change ranges from a maximum circulation of 6.6 min air change for Building H to a 17.9 min air change for Building A.

The cubic volume per ton of calculated refrigeration load ranges from a minimum of 2880 cu ft per ton to a maximum of 6050 cu ft per ton in Buildings B and I respectively. Thus, the windowless office building, Building I which would not have been possible without air conditioning, indicates the reduction in first cost for air conditioning equipment if the building factor is controlled. A reduction in operating cost per cubic foot of conditioned space would likewise result with an insulated building of the windowless type such as Building I. A variation in floor area per ton ranges from 217 sq ft per ton in Building B up to 574 sq ft per ton in Building D.

Conclusions

The analyses and summaries herein presented are descriptive of building air conditioning installations in ten buildings located throughout the United States.

No two of these buildings were designed in exactly the same manner. It is possible by making use of the various alternate methods of installing an air conditioning system in an office building to make such an installation in many different ways, when it is considered that there are many types of equipment available with many alternates which can be offered for accomplishing a definite specified and guaranteed result.

The variables in application factors which are beyond control of the designing engineer are numerous and it would thus be futile to specify a given set of rigid standards to govern the installation of air conditioning systems in office buildings.

It does appear to be practical, however, to compare the overall average composite data selected from a large group of representative large sized installations to establish the limits which have been successfully applied in experience to produce for the purchasing public a satisfactory result in air conditioning.

With the many types of air conditioning apparatus available, and the numerous application factors encountered, the air conditioning engineer will find the office building air conditioning system a test of ingenuity and resourcefulness if the most economical system is to be selected for each and every installation.

A. S. T. M. Committee on Thermal Insulating Materials

As a result of discussions over several years concerning the desirability of undertaking work in the field of thermal insulating materials, the *American Society for Testing Materials* has organized a new standing committee designated C-16 on Thermal Insulating Materials.

For some years *A.S.T.M.* Committee C-8 on Refractories has had important standardization and research work under way involving materials generally considered in the field of high-temperature insulations. The new Committee C-16 will undertake work on insulations for use at somewhat lower temperatures, as for example, in the piping field and similar type industrial insulations.

For the time being it is not proposed to include the housing insulations, although a number of tests applicable to the industrial insulations will no doubt also be of interest in connection with other types of insulation.

At the organization meeting on March 10 the committee discussed the possibility of developing tests for the following properties: handleability, hardness, abrasive resistance, modulus of rupture, compressive strength, loss of weight, volume shrinkage, shrinkage on drying (plastic materials), slump (plastic materials), coverage (plastic materials), adhesiveness (plastic materials), surface finish (plastic materials), and vibratory tests.

Mention was also made of the various thermal properties of interest in addition to thermal conductivity, namely, properties such as specific heat, coefficient of expansion, etc. The committee is working in close cooperation with the American Society of Heating and Ventilating Engineers, particularly with respect to the determination of thermal conductivity.

Meeting in the Mountains

The Homestead . . . . . Hot Springs, Va. . . . . June 20-22

HIGH in the hills of historic Virginia, A.S.H.V.E. members will find the 1938 Summer Meeting place a picturesque spot that is cool, invigorating and famed for its scenic beauty. The Homestead, where the meeting will be held June 20 to 22, provides every facility for rest or recreation and is accessible from all parts of the United States by railroad or automobile.

The scenic and historic beauties of Virginia are well known and motorists from the north, east and south can enjoy the new Skyline Highway enroute to Hot Springs. Others may want to make side trips to Natural Bridge, the Caverns, Monticello or Williamsburg.

A technical program of unusual interest has been prepared and the Committee on Arrangements plans a varied program of entertainment for members and ladies.

Service at the Homestead is on the American Plan and the daily rate includes room and bath with three meals. There will be no extra charge for the banquet to guests of the Homestead.

Plans should be made now to come to Hot Springs June 20 to 22 where a royal welcome awaits everyone. Society members in Virginia, West Virginia and the Officers of the Atlanta, Washington and Cincinnati Chapters will have charge of the arrangements. The Committee on Arrangements: J. A. Donnelly, Chairman, Largent; I. B. Helburn, Cincinnati; R. M. Johnston, Blacksburg; E. W. Klein, Atlanta; L. F. Nordine, Washington, D. C.; J. K. Peebles, Charlottesville; J. A. Shanklin, Charleston.

TENTATIVE PROGRAM

(All Events Eastern Standard Time)

Sunday, June 19

2:00 p.m. Meeting of the Council

Monday, June 20

8:30 a.m. Registration

9:30 a.m. TECHNICAL SESSION:

- A Test Method for Air Cleaning, by R. S. Dill
- The Flow of Air Through Exhaust Grilles, by A. M. Greene, Jr., and M. H. Dean
- The Condensation Nuclei Content of the Air as Related to Air Freshness, by R. A. Nielsen

11:30 a.m. Concert for Ladies

2:00 p.m. Golf Tournament (Homestead Course) (18 Hole Kickers Handicap)

2:30 p.m. Tennis Tournament

2:30 p.m. Carriage Ride (Skyline Drive)

8:00 p.m. Movies

8:30 p.m. Bridge Party

10:00 p.m. Dancing (Crystal Room)

Tuesday, June 21

9:30 a.m. TECHNICAL SESSION:

- Heat Transfer Through Single and Double Glazing, by M. L. Carr, R. A. Miller, Leighton Orr and Alan C. Byers
- Glass Heat Transfer Coefficients, by D. Shore
- Performance of Surface Coil Dehumidifiers for Comfort Air Conditioning, by G. L. Tuve and L. T. Seigel

10:30 a.m. Ladies Golf and Putting Contest

2:00 p.m. Golf Tournament (Cascade Course) (18 Hole Medal Play) (Research and Chapter Cups)

2:00 p.m. Motor Trip and Inspection of Covington Paper Mill

2:30 p.m. Ladies Bridge and Tea

7:00 p.m. Semi-Annual Banquet and Dance

Wednesday, June 22

9:30 a.m. TECHNICAL SESSION:

- Seasonal Variations in Effective Temperature Requirements, by F. E. Giesecke, W. H. Badgett and F. C. Houghten
- Cooling Requirements for Summer Comfort Air Conditioning in Toronto, by C. Tasker
- The General Reactions of 275 Workers to Summer Cooling and Air Conditioning, by A. B. Newton and F. C. Houghten
- Study of Summer Cooling in the Research Residence Using a Small Capacity Mechanical Condensing Unit, by A. P. Kratz, S. Konzo, M. K. Fahnestock and E. L. Broderick

2:00 p.m. Golf Tournament

Pres. Gurney Addresses Chapters

Pres. E. Holt Gurney, Toronto, has returned recently from an extended trip which took him to the Pacific Coast on a visit to a number of A.S.H.V.E. Chapters. The talk which he gave showed how Air Conditioning Brings Human Problems to the Engineer and he had numerous speaking engagements in the following cities.

His first stop was in St. Louis, where he addressed a meeting of the St. Louis Chapter on February 28. In New Orleans in addition to a luncheon meeting, on March 3 President Gurney spoke at Tulane University to a group which included local members of the A.S.H.V.E., the *Louisiana Engineering Society*, the New Orleans Public Service, Inc., and members of the faculties and engineering students at Tulane and Louisiana State Universities. On March 5 a meeting of the Texas Chapter was held at Houston, with a record attendance for the Chapter.

Traveling to the coast, he visited Los Angeles and the Southern California Chapter where he was the featured speaker at a meeting on March 14. At Berkeley President Gurney gave a lecture before the College of Engineering, of the University of California on March 16, where the students were particularly interested in the work of the Society. Members of the Society's Golden Gate Chapter at San Francisco entertained President Gurney as their guest on March 18 and on this occasion he had the pleasure of presenting them with their charter. Engineering students and members of the faculty of Oregon State College, Corvallis, heard an address by President Gurney on March 21, when he was able to visit them during part of the day. On March 22 he spoke to the Pacific Northwest Chapter of the A.S.H.V.E. in Seattle, Wash., and on April 12 he visited the Manitoba Chapter at Winnipeg.

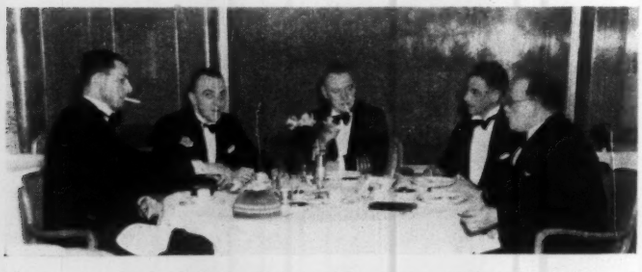
President Gurney was accompanied on the trip by Mrs. Gurney and they have both expressed their appreciation of the hospitality and friendliness extended to them wherever they visited.

After a few days in Toronto President Gurney's schedule calls for him to address the Western New York Chapter on April 25, following a Council meeting in Buffalo, and to visit the Michigan Chapter at Detroit on April 28, the Illinois Chapter at Chicago on May 9, and the Wisconsin Chapter at Milwaukee on May 10.

W. L. Fleisher Visits Europe

On April 4, W. L. Fleisher, chairman of Committee on Research, returned to New York on the Queen Mary after an extended trip to England, Belgium and France, where he had the opportunity to interview leading authorities in the political and scientific fields.

Soon after his arrival in London, he met Thomas Chester and attended a meeting of the *Institution of Heating and Ventilating Engineers* on March 2, where Mr. Fleisher outlined the research activities of the A. S. H. V. E. before 100 members present. At this time he emphasized the importance of cooperation between the two organizations in correlating any research which might be of mutual interest.



Seated at the Chief Engineer's Dinner on the Queen Mary: (*left to right*) L. J. Fowler, London; C. H. Crabtree, Bradford; W. Sutcliffe, staff chief engineer, Queen Mary; W. L. Fleisher, New York, chairman, Committee on Research; and W. G. Case, London

Climatic conditions in England were observed as being much different than in America, and it was apparent to Mr. Fleisher that the Comfort Chart would undoubtedly have to be modified to satisfy English conditions. There was some evidence presented to him which indicated that the effective temperature line might be approximately 68-69 deg ET as the optimum summer condition for London. It is expected that current research in this field will be developed in the near future which will make it possible to study the reactions of the English people to various air conditioning environments.

On the 11th of March, Mr. Fleisher made a speech before the *Societe des Ingenieurs et Industriels* of Belgium, which was delivered in English and later translated into French for reproduction in their engineering proceedings.

A conference was arranged with Prof. André Missenard in Paris on March 15, and Mr. Fleisher reports a very interesting discussion with him on research problems connected with comfort and health on the continent. It seemed that Professor Missenard was especially interested in the progress which we were making in applying air conditioning to the treatment of disease and especially its use in fever therapy.

Before returning to the United States, Mr. Fleisher spent some time at Cambridge University where he delivered a lecture before the engineering students under the direction of Professor Inglis. This discussion dealt primarily with an outline of air conditioning and its particular opportunity in England as a career profession.

The *Institution of Heating and Ventilating Engineers* was sufficiently interested in his brief and extemporaneous talk on March 2 to ask him to give a complete lecture on Air Conditioning with Reference to Effective Temperature, which he delivered on March 28 before about 200 members and invited guests, including members of the *English Gas Association*. Although his talk was of a very simple character, the questions brought out by it covered the whole field of research and should be of great interest to the American Society of Heating and Ventilating Engineers when they are published in the *Journal* of the *Institution of Heating and Ventilating Engineers*, which is to be done shortly.

Mr. Fleisher had great opportunity while in England and on the Continent to discuss gas defense and the conditioning of underground storage spaces not only for the storage of ammunitions, but also for the safety and accommodation of the people. The greatest interest in England today and the greatest opportunity for air conditioning installation, he states, lies in this particular field.

Appointment of Public Health Committee

Pres. E. Holt Gurney has announced the selection of the personnel for a Committee on Engineering In Its Relation to Public Health which consists of J. J. Aeberly, chairman, W. H. Carrier, Philip Drinker, E. V. Hill, John Howatt, A. C. Willard and B. M. Woods.

The purpose of this committee is to serve the fields of engineering and public health and bring about a closer relationship and coordination of comfort between the doctor and engineer.

This group is also to cooperate with the Society's Committee on Research in preparing a standard method of procedure for handling all matters of Society work dealing with health.

On June 6-9, 1938, the 23rd annual meeting of the *American Association of Industrial Physicians and Surgeons* will be held concurrently with the second annual Midwest Conference on Occupational Diseases at the Palmer House, Chicago, Ill. The program is of interest to engineers, industrialists, hygienists, and chemists, as well as physicians and surgeons.

Educational Forum

Beginning on April 12 six weekly forum meetings will be held in Washington, D. C. every Tuesday evening where the latest

thoughts and recommendations of experts in the field of heating, ventilating and air conditioning will be presented. The list of speakers includes E. N. Foss, M. P. Isaminger, W. R. Lockhart, H. Peacock, D. M. Robinson, F. E. Spurney, and T. H. Urdahl.

The forum is intended for owners, engineers and operators or anyone having any degree of responsibility in providing healthful and comfortable conditions for public assembly. The project is being sponsored by the Health Department of the District of Columbia in conjunction with an Industry Advisory Committee which is composed of C. F. Browning, chairman, Health Department; Dr. C. C. Dauer, Health Department, W. B. Henderson, *Air Conditioning Manufacturers' Association*, E. V. Fineran, Washington Gas Light Co., R. H. Lucas, office of the Plumbing Inspector, L. Ourusoff, president, Washington, D. C. Chapter, American Society of Heating and Ventilating Engineers, S. L. Gregg, Potomac Electric Power Co., and J. S. Bartlett, Electric Institute of Washington.

F. W. Morse Elected Vice-President

Announcement has been made of the election of F. W. Morse as vice-president and director of the Chamberlin Metal Weather Strip Co., Detroit, Mich., in recognition of his ability and service rendered the company.

Northern Ohio Chapter

January 12, 1938. The meeting of Northern Ohio Chapter held at the Cleveland Club was attended by 92 members and guests, 48 being present for dinner, according to the report of Secy. C. A. McKeeman. Following the reading and acceptance of the minutes, reports were received from the treasurer and the Entertainment Committee.

L. T. Avery reported favorably for the Standards Committee on the Code of Minimum Requirements for Comfort Air Conditioning to the effect that it was a step in the right direction and should have the support of Society members.

Mr. Avery also reported briefly on the status of the movement to bring the 1940 Annual Meeting of the Society and Exposition to Cleveland.

A communication was read from M. F. Rather that his moving to New York would make it necessary for him to retire from Northern Ohio Chapter activities and he expressed his regret at breaking off the former pleasant associations. Mr. Avery then voiced the appreciation of the Chapter for the interest and active part taken by Mr. Rather since 1918. A motion was made and passed that: Max Rather be given Godspeed by the Chapter as an expression of appreciation for his work in the Northern Ohio Chapter.

The names of P. D. Gayman and H. P. Rankine, Jr. were presented to the Chapter and duly elected.

The speaker of the evening was Prof. C.-E. A. Winslow, New Haven, Conn., who gave a very interesting talk on the Effect of Radiation on the Human Body, followed by an active question period.

Wisconsin Chapter

February 21, 1938. A dinner and meeting of Wisconsin Chapter was held at the City Club, and C. H. Randolph introduced the coffee speaker, D. J. McNally, Better Business Bureau, Milwaukee Association of Commerce. Mr. McNally told of schemes employed by merchants in their advertising and advised of methods used to combat fraudulent advertising with warnings and court orders.

Pres. J. H. Volk held a short business meeting in which the new amendment for the change in the election of the Board of Governors was discussed, voted upon and passed.

The speaker of the evening, F. R. Bichowsky, Dow Chemical Co., Ann Arbor, Mich., gave a talk on Air Conditioning with Lithium Chloride, presenting the history of dehumidification without refrigeration from its earliest stages, together with the various methods employed and the chemicals used in such work. Mr. Bichowsky explained the difference from the usual silica-gel and activated alumina type of dehumidification with that of continuous process used in the lithium chloride method. He further explained the simplicity of control with lithium chloride by regulating its concentration.

Secy. H. C. Frentzel reports that the talk was enlightening and technical in nature and was of interest to the members and guests.

Cincinnati Chapter

February 8, 1938. On this date members of the Cincinnati Chapter held their regular meeting in Middletown, Ohio. In the afternoon 18 were conducted on a tour through the plant of the American Rolling Mill Co. to view the manufacturing of sheet products. At 6:30 p. m. dinner was served to 40 members at the Manchester Hotel, Middletown, and the meeting which followed was presided over by Pres. I. B. Helburn.

Announcement was made regarding the annual meeting of the Technical Scientific Societies Council of the City of Cincinnati, February 18, at Cincinnati, with Miller McClintock, director of Bureau of Street & Traffic Research, Harvard University, as speaker on the subject of Traffic in the City of Tomorrow.

President Helburn then introduced Dr. Anson Hayes, director of research, American Rolling Mill Co., who gave an address on Research Activities in the Steel Industry. Dr. Hayes told of the necessity for and the problems of research in the steel industry in order to provide the requirements for the necessities of life, namely, food, clothing, shelter, transportation. After this interesting address, the meeting adjourned to the new research building, where, following an inspection trip, an enthusiastic vote of thanks was given to Dr. Hayes, Tom Byrd and many others who helped to provide an interesting and enjoyable day and evening, according to the report of Secy. H. E. Sproull.

New York Chapter

March 21, 1938. The design of a heating plant for a small house was the subject of the meeting of the New York Chapter held at the Building Trades Club, New York City, when 70 members and guests attended, and Pres. W. E. Heibel presided. Plans for a house were sent to members of the Chapter before the meeting, giving dimensions and heat losses which totaled 70,000 Btu per hour, and the members were asked to come prepared to discuss what type heating system should be installed.

Three speakers had been previously selected to present detailed plans for heating. M. C. Barnum showed his plans for the house which included a steel warm-air oil-burning furnace complete with humidifier and air filters. R. L. Stinard presented plans for heating the house with a one-pipe steam system, including a humidifying device in which the moist air was introduced into the living room. A similar plan of humidifying was used by T. W. Reynolds, secretary of the Chapter, in plans he presented for heating the house with a forced hot water plant.

Following the formal presentation of the three plans, the meeting was thrown open for a discussion which was lively and participated in by members and guests.

Announcement was made that at the last meeting of the season in May the speaker would be a representative of the Federal Bureau of Investigation.

February 21, 1938. Steam jet refrigeration as applied to air conditioning was the subject of the regular monthly meeting of the New York Chapter, held at the Building Trades Employers Club. The speakers of the evening were A. R. Mumford and A. A. Markson, research engineer and assistant research engineer, respectively, of the New York Steam Corp.

The speakers showed that if steam jet refrigeration for air conditioning were applied to 10 per cent of the space now served by district steam in New York, the local utility's load factor or ratio of the average month to the peak month would increase from 47.35 per cent to 51.6 per cent, while if 20 per cent of the space were conditioned, the annual load factor would be 56.0 per cent. The purpose of the speakers' talk was to demonstrate

and discuss air conditioning applications which fall into the economical range of water vapor refrigeration; to show that steam jet refrigeration should be treated as a direct process in order to obtain real operating advantages in economies; to familiarize engineers with the load characteristics of steam jet apparatus, and to discuss condenser water from a constructive engineering viewpoint.

They analyzed steam jet refrigeration and explored its limitations theoretically and then applied some of their findings to steam jet installations operating in New York City. They pointed out that where central station steam is available in Manhattan the operating costs of steam jet are such that it cannot be disregarded in any economic set-up.

Following their talk, Charles Hoffman, chief engineer, First National Bank Bldg., New York, spoke regarding some of his operating experiences where steam jet equipment is being used. O. L. Maddux, a visitor from Hamilton, Ont., discussed his findings and pointed out that the observed results agreed with the work done at the A. S. H. V. E. Research Laboratory in Pittsburgh.

Western Michigan Chapter

March 30, 1938. The regular monthly meeting of Western Michigan Chapter was held at Michigan State College, East Lansing, in conjunction with the seventh annual session of the Forced Warm Air Conference. At the banquet in the ballroom of the Union Building about 250 members of the Western Michigan Chapter and conference attendants were served, and among this number were 15 or 20 members of the Michigan Chapter of the A. S. H. V. E.

The speaker of the evening was J. F. McIntire, Detroit, Mich., 1st vice-president of the A. S. H. V. E., whose subject was Research and Air Conditioning—Magic Words.

Short informal talks were made by Prof. J. D. Hoffman, West Lafayette, Ind., chairman of the Code Committee of the *National Warm Air Heating and Air Conditioning Association,* and L. R. Taylor, president of the Association.

In view of the special character of the meeting, all routine business of the Western Michigan Chapter was dispensed with until April.

February 14, 1938. At a meeting of the Western Michigan Chapter at the Mertens Hotel, Grand Rapids, 52 were present for dinner and 15 additional members and guests attended the business session. Reports of the Secretary, Treasurer, and Chapter delegate were read and approved.

As new business, Pres. W. W. Bradfield brought up two matters relating to the activity of the Chapter. It was decided to have a Constitution and By-Laws Committee appointed for revision of the present Constitution. A motion for the appointment of a local committee for consideration of minimum requirements for heating, ventilating and air conditioning systems was approved.

It was announced that in March members of the Michigan Chapter and those in attendance at the Michigan State College Short Course would be guests of Western Michigan Chapter, at a meeting held in East Lansing.

A number of the technical papers presented at the Society's Annual Meeting were abstracted for the benefit of those who could not be present.

O. D. Marshall, chairman of the Program Committee, introduced B. F. McLouth, who discussed the report of research studies at the University of Illinois, Study of Methods of Control and Types of Registers as Affecting Temperature Variations in the Research Residence.

S. H. Downs then discussed the paper reporting results of tests at the John B. Pierce Laboratory of Hygiene, New Haven, Conn., Physiological Reactions and Sensations of Pleasantness Under Varying Atmospheric Conditions.

The paper, Condensation Within Walls, giving the results of tests conducted at the University of Minnesota, was presented by Prof. L. G. Miller.

Mr. Marshall summarized the paper, Cooling Tower Equipment and Its Relation to Water Conservation.

President Bradfield ably gave his observations on the new Lincoln Vehicular Tunnel as gathered from his inspection of it during the recent Annual Meeting. He referred largely to the ventilating aspects of the tunnel and maintained the interest of the members until the meeting adjourned at 10:30 p. m.

Washington, D. C. Chapter

March 9, 1938. This meeting of the Washington, D. C. Chapter was a joint session with the local section of the *American Society of Mechanical Engineers* and was held in the Potomac Electric Power Co.'s auditorium. The meeting was largely attended and approximately 50 A. S. H. V. E. members were present.

The routine business of the Chapter was dispensed with and H. G. Thielscher presided.

Preceding the lecture, Pres. L. S. Ourusoff made a few brief announcements of interest to the Chapter.

Immediately following, Mr. Thielscher introduced V. A. Schlenker, consulting engineer and lecturer at Columbia University, New York, whose subject was Noise, Sounds and Vibrations in Ventilating Ducts. The address was instructive and its interest was added to by demonstrations of equipment.

Secy. L. F. Nordine reports that the meeting adjourned at 10:30 p. m.

Minnesota Chapter

March 14, 1938. Pres. R. E. Backstrom called the meeting of Minnesota Chapter to order following dinner at the Minnesota Union.

Prof. F. B. Rowley spoke with reference to facilities available at the Institute Center, University of Minnesota, and told of organizations that have used the accommodations for conferences of three or four days' duration. After discussion President Backstrom appointed the following members as a committee to take action relative to this matter: Professor Rowley, E. F. Jones, H. M. Betts, H. S. Morton and R. E. Backstrom.

A. J. Huch, chairman of the Entertainment Committee, reported with regard to the May 16 party at Golden Valley Golf Club. He announced plans, which included an all day session, with golf for the men, golf and cards for the ladies, and a dinner and dance in the evening. Transportation facilities have been arranged for all who desire to attend.

President Backstrom then turned the meeting over to Mr. Morton who introduced the speaker of the evening, J. F. Murphy, General Electric Co., Chicago, who gave an interesting address on the Trend of Air Conditioning.

Approximately 55 members and guests attended according to the report of M. H. Bjerken, acting secretary.

February 14, 1938. The regular meeting of the Minnesota Chapter was held at the Minnesota Union, University of Minnesota with approximately 100 members and guests attending. Dinner was served at 6:30 p. m., after which Pres. R. E. Backstrom called the meeting to order.

Secy. F. C. Winterer read two notices which were received from Secy. A. V. Hutchinson with reference to distribution of The Heating, Ventilating, Air Conditioning Guide 1938 and the new dues rate effective in 1938 for Student Members.

President Backstrom outlined briefly his experience at the 44th Annual Meeting of the A. S. H. V. E. in New York, mentioning the names of approximately 20 or 25 Minnesota Chapter members who attended, and covering activities and the conference of Chapter delegates.

A card with greetings from Mr. and Mrs. Albert Buenger was read by President Backstrom and received by those in attendance with great pleasure.

N. D. Adams, a member of the Society's Council, spoke of

the research work of the A. S. H. V. E. and the Speakers Bureau, and also gave the membership status. Mr. Adams then presented an interesting and educational resumé of his trip from New York to Bermuda and return, which included high lights on climate, scenery, mode of living, etc. encountered during the trip.

The meeting was then turned over to the speaker of the evening, Prof. F. B. Rowley, University of Minnesota. The subject of Professor Rowley's address was Condensation Within Walls. He outlined the moisture problems that have come up on account of high relative humidities due to the marked advances in the science of air conditioning. With tabular data, illustrations of test houses, sections of walls and ceilings, he explained the moisture gain within walls and ceilings as a result of tests made of various constructions and at various per cents of relative humidity and temperature. After Professor Rowley's talk, numerous questions were asked, which indicated that the subject was of great interest and that those in attendance had derived some real value from the meeting.

Pittsburgh Chapter

March 14, 1938. Pres. J. F. Collins, Jr. called the Pittsburgh Chapter meeting to order in the private dining room of Stouffer's Restaurant. Secy. T. F. Rockwell read the minutes of the February meeting which were approved.

L. S. Maehling, treasurer, reported on the Chapter's financial condition.

Other committee chairmen submitted reports as follows: Program Committee—E. C. Smyers; Membership Committee—P. A. Edwards; Air Conditioning Committee—G. G. Waters; Convention Committee—R. A. Miller; and Social Committee—R. J. J. Tennant.

The chairman of the Student Committee, Professor Rockwell, introduced the first speaker, Bruce Jacobi, senior, Carnegie Institute of Technology, who spoke briefly on an Outlook for Air Conditioning, in which he abstracted material appearing in a newspaper article.

President Collins then introduced Prof. B. M. Woods, University of California, Berkeley, Calif., who spoke on Human Factors in Air Conditioning. In his paper Professor Woods expressed the idea that mechanical equipment had advanced to the point where any reasonable combination of atmospheric conditions in an enclosure could be produced and controlled, but just what combinations of conditions might be most beneficial to human beings still remains to be determined. Professor Woods then referred to various investigations being made by medical men in this field and discussed the results to date. He concluded his presentation by exhibiting a number of slides, some of which were pictures of simple cooling devices used in the hot dry climate of the Southwest.

This paper proved to be of unusual interest to the 45 members and guests present.

February 14, 1938. Pres. J. F. Collins, Jr. called the meeting of Pittsburgh Chapter to order at 7:35 p. m. in the private dining room of Stouffer's Restaurant. Secy. T. F. Rockwell read the minutes of the January meeting, which were approved.

The following committee chairmen reported on the activity of their various committees: National Activities—Prof. C. M. Humphreys; Research—F. C. McIntosh; Student Activities—Professor Rockwell; Social Committee—R. J. J. Tennant. As part of the committee's work, H. A. Canon, member of the Social Committee, related some anecdotes about Abraham Lincoln.

President Collins read a report of the conference of Chapter delegates during the Society's Annual Meeting in New York.

T. E. Purcell, superintendent, Allegheny County Steam Heating Co., was introduced by President Collins as the principal speaker. Mr. Purcell's subject was District Heating, and he began by reviewing the history of the industry, going back to the original plant in Lockport, N. Y. He then described the beginning of the company with which he is associated, emphasizing the difficult and expensive construction of the distribution system required by the local topography. The paper was concluded with a discussion of the economics of district heating, including such items as load factor, use factor and optimum line pressure.

The 54 members and guests greatly enjoyed Mr. Purcell's presentation and an interesting discussion followed.

Massachusetts Chapter

March 15, 1938. Sixty members and guests attended the meeting at the Massachusetts Institute of Technology, when supper was served in the Walker Memorial cafeteria preceding the business session of Massachusetts Chapter.

Pres. James Holt presided and announced the following three nominations for the Board of Governors for the ensuing year to be voted on after the April meeting: D. S. Boyden, H. W. Frederick and Alfred Kellogg.

The subject of discussion for the evening was Refrigeration in Air Conditioning. Different types of refrigeration systems were discussed by two speakers. W. E. R. Karsten, General Electric Co., Bloomfield, N. J., spoke on compression systems, and E. L. Champion, Ingersoll-Rand Co., New York, spoke on steam jet and centrifugal water vapor systems.

According to Secy. H. C. Moore a general discussion followed the meeting.

February 15, 1938. Eighty members and guests of Massachusetts Chapter held a meeting at the Massachusetts Institute of Technology and preceded the technical session with supper at the Walker Memorial Cafeteria. Early in the evening Pres. James Holt gave a brief summary of the most important papers presented at the Society's Annual Meeting held in New York. The retiring national president was then called upon for brief remarks and Col. D. S. Boyden responded with general Society statistics and his appreciation of the support given to him by Massachusetts Chapter.

Three speakers were introduced and spoke on the general topic of Recent Advances in Temperature and Humidity Controls: J. H. Barrett, who described in detail a recent local installation; T. F. McCoy gave a description of general developments in this field and J. H. Colby told of experiences with different thermostatic elements and humidity controls, especially in mills.

Secy. H. C. Moore reports that many questions were answered by the three speakers when the meeting was open for general discussion.

Illinois Chapter

March 14, 1938. The March meeting of the Illinois Chapter was the largest of the current season and was held at the Brevoort Hotel, Chicago, with 118 present at dinner and 185 in attendance at the meeting. The meeting was called to order at 7:55 p. m. by Pres. S. I. Rottmayer.

New Chapter members, M. J. Stevenson, O. G. Ward, H. J. Prebensen and F. W. Wills, were introduced by E. M. Mittendorff, Membership Committee chairman.

Tom Brown, chairman of the Meetings Committee, announced that Prof. S. Konzo, University of Illinois, was scheduled as speaker at the April meeting on Forced Air Heating. He also told of plans for the May meeting with golf in the day time and a combined business and social meeting in the evening.

G. H. Blanding introduced the speaker of the evening, Prof. G. L. Larson, University of Wisconsin, Madison, Wis., past president of the A.S.H.V.E. Professor Larson's subject was Mechanical Hot Water Heating Systems. He gave a general review of hot water as a heating medium, traced the development of the forced circulation system, compared it with the gravity method and presented various arrangements of piping and other fundamental phases of the subject. His talk called for considerable discussion entered into by Chapter members.

At the conclusion of the meeting a rising vote of thanks expressed the Chapter's appreciation of Professor Larson's visit.

Announcement of the election of the following as members of the Nominating Committee was made by the tellers: J. J. Aeberly, Tom Brown, J. J. Hayes, John Howatt, C. W. Johnson, J. H. Milliken and C. E. Price.

February 14, 1938. Pres. S. I. Rottmayer presided at a meeting of the Illinois Chapter, held at the Brevoort Hotel, Chicago, with 82 members and guests present. The minutes of the January meeting were read by Secy. C. E. Price after which E. M. Mittendorff introduced two new Chapter members—J. H. Manny and C. E. Beery.

Secretary Price gave a detailed report of the 44th Annual Meeting of the A. S. H. V. E. and other activities of the week in New York. J. J. Aeberly gave a brief summary of the meetings held by the Council while in New York, and he also explained the fundamentals of the health work in relation to air conditioning which the Society is undertaking.

President Rottmayer then introduced J. T. W. Babcock, engineer, *Western Factory Insurance Association,* who spoke on the subject of Fire Hazards in Connection with Heating, Ventilating and Air Conditioning. He pointed out that it is a matter of history that almost every engineering development is accompanied with its era of fires and that air conditioning has presented new problems which must be faced. He stated that safeguards begin on the drawing board of the designer and urged the cooperation of the engineers in eliminating the hazards which might be set up by improperly designed systems.

Mr. Babcock was followed by R. E. Maginnis, special agent, American District Telegraph Co., who described Methods of Protection Against Fires.

It was apparent from the two talks that insurance companies are giving careful study to air conditioning and that the engineers should know what regulations are being prepared and should cooperate to the best interests of all concerned.

After discussion by Mr. Aeberly, R. E. Hattis, E. P. Heckel and A. G. Sutcliffe, the Secretary was instructed to write to the organizations of the two speakers represented and thank them for making their instructive messages available to the Chapter.

Manitoba Chapter

March 24, 1938. The meeting of the Manitoba Chapter at the Fort Garry Hotel was presided over by Pres. D. F. Michie and attended by 15 members and guests, while 30 were present for the technical discussion. Roll call was taken and the minutes of the February meeting were approved.

President Michie read a letter which had been sent to the Minnesota Chapter, inviting them to a joint meeting in Winnipeg on May 28.

The following members of the Nominating Committee were named by President Michie to report at the next meeting: J. B. Steele, chairman, Frank Thompson and P. L. Charles.

Welcome was extended to R. P. Hinton, a new Chapter member.

The meeting adjourned to Theater A, Manitoba University, where the guest speaker, D. M. Forfar, Grinnell Co., Minneapolis, gave an interesting address on Mechanical Systems of Hot Water Heating.

A vote of thanks to Mr. Forfar on behalf of the Chapter was moved by Mr. Steele, for his interesting address.

February 24, 1938. The Manitoba Chapter met at the Fort Garry Hotel, Winnipeg, at 6:30 p. m., with Pres. D. F. Michie presiding and 14 members and guests present.

A letter was read from Pres. E. Holt Gurney of the A. S. H. V. E. regarding his proposed visit to the Chapter in April. A letter from Secy. A. V. Hutchinson of the Society was read, and stated the new rate of dues for Student Members.

Frank Thompson spoke about the possibility of inviting the Minnesota Chapter to Winnipeg and President Michie welcomed three new members, Theodore Kipp, P. L. Charles and Ivan McDonald.

M. W. Shears of the Ontario Chapter was called on and made a few remarks regarding their membership campaign.

The guest speaker of the evening was W. D. Hurst, City of Winnipeg's Engineer's Department, who gave an interesting paper on Treatment of Sewage as practiced at the new modern sewage disposal plant in Winnipeg. After the lecture there was a general discussion.

A vote of thanks was given to Mr. Hurst on behalf of the Chapter, before adjournment at 9:45 p. m.

January 15, 1938. Pres. D. F. Michie presided at a meeting of Manitoba Chapter in the Fort Garry Hotel, Winnipeg, at 6:30 p. m. B. F. McLouth, Dail Steel Products Co., Lansing, Mich., was guest speaker of the evening and conveyed greetings from both the Michigan and Minnesota Chapters of the Society.

Following dinner the meeting adjourned to Manitoba University, where a public meeting was held with about 60 members and guests listening to an interesting address on Engineering Sells Mechanical Warm Air Systems.

Western New York Chapter

March 14, 1938. About 88 members and guests were present at the dinner and meeting of the Western New York Chapter held at the Iroquois Gas Co., Buffalo, where they were welcomed by O. F. Flumerfelt.

The minutes of the February meeting were read and approved. L. P. Saunders, chairman of the Membership Committee, reported that 12 new members had been added to the Society from Western New York.

Pres. B. C. Candee reported in regard to the McNaboe Bill and thanked the members for their efforts in writing their representatives. President Candee then read Senate Bill No. 174 together with an extract from the report of the A. S. H. V. E. Council Committee on this bill, and, after considerable discussion, it was voted to defer action until the April meeting.

A brief report of the business transacted at the 44th Annual Meeting of the Society was given by President Candee, who announced that he had been appointed a member of the Speakers Bureau Committee for 1938 and requested assistance and suggestions from the local membership.

The meeting was then turned over to H. C. Schafer, who introduced W. B. Garrison, Jr., chief engineer, American Gas Products Corp. Mr. Garrison told briefly of some of the interesting highlights in connection with supplying gas for the New York World's Fair.

After a rising vote of thanks to their hosts, the meeting adjourned at 9:00 p. m., according to the report of Secy. W. R. Heath.

February 8, 1938. Approximately 30 members and guests of Western New York Chapter met at the University Club, Buffalo, according to the report of Secy. W. R. Heath. The minutes of the January meeting were read and approved and it was voted to postpone the date of the April meeting until later in the month to suit the convenience of J. F. McIntire, 1st vice-president of the A.S.H.V.E.

Pres. B. C. Candee read the McNaboe Bill and Governor Lehman's veto message of June 6, 1937. He then read a resolution, which after some discussion was unanimously adopted. President Candee urged the members to personally communicate with their assemblymen and state senators, requesting them to vote against the bill when it is brought up again in the near future.

M. C. Beman introduced the speaker of the evening, Prof. Philip Drinker, Harvard School of Public Health, Boston, Mass., who spoke on Atmospheric Pollution and Dust Control. Professor Drinker covered the most common forms of industrial pollution of air and spoke of the various methods available for counteracting such pollution. During the discussion Theodore Hatch, dust control engineer, New York State Department of Labor, told of some of his experiences in similar work.

Atlanta Chapter

April 5, 1938. The Atlanta Chapter met at the Atlanta Biltmore Hotel at 8:00 p. m. for a meeting which was attended by a number of visitors from other engineering societies, who had been invited in order that they might hear the address of Albert Buenger on Reducing the Cost of Air Conditioning.

The usual Chapter business was dispensed with, and after announcements by Secy. C. T. Baker, Pres. E. W. Klein introduced Mr. Buenger, Dayton, Ohio, formerly a member of the A.S.H.V.E. Council and chairman of the Guide Publication Committee. Mr. Buenger delivered a timely and interesting address on his subject and presented a wealth of facts on the economies of air conditioning, including a discussion of types of plants best suited to meet specific conditions. He stressed the need for a careful and exact analysis of all proposed air conditioning installations and emphasized the necessity for thoroughness in developing all information possible before deciding on the type of plant and general layout to meet any given situation. The speaker gave some interesting cost statistics and showed how operating costs can be materially reduced by the use of properly applied insulation, the use of awnings over windows, and the treatment of roof surfaces to reduce sun effect. One interesting typical example was developed in detail which concerned the operating cost of an air conditioning plant in a ladies' ready-to-wear shop.

The attendance was unusually large and all exhibited a keen interest in the topic covered and in the discussion which followed.

Plans were made for a special meeting of the Chapter on April 20 to hear Prof. F. E. Giesecke, 2nd vice-president of the A.S.H.V.E.

It was decided to hold in abeyance the question of the election of officers until the May meeting.

March 1, 1938. Pres. E. W. Klein presided at the regular meeting of the Atlanta Chapter at the Biltmore Hotel. Following the reading of the minutes and communications received by Secy. C. T. Baker, President Klein appointed a Nominating Committee to nominate officers for the coming year and present the slate at the April meeting for action. The following members were appointed on the Nominating Committee: L. F. Kent, *chairman*, M. G. Driscoll, Leo Sudderth, Jr., C. L. Templin and W. J. McKinney.

Following the conclusion of the usual order of business, President Klein introduced M. G. Driscoll, Bryant Equipment Co., who gave an interesting and instructive talk on The Properties and Uses of Silica Gel. The facts presented by Mr. Driscoll, who was thoroughly conversant with the subject, developed an interesting discussion and every one present showed a keen interest in the data presented.

Texas Chapter

March 5, 1938. An outstanding meeting of the Texas Chapter was held at the Rice Hotel, Houston, when 71 members and guests gathered from College Station, Dallas, Chicago, Kansas City, Philadelphia, Port Arthur, Kingsville, Beaumont, Galveston, New Orleans and Houston for dinner with the president of the A.S.H.V.E., E. Holt Gurney, Toronto, Canada.

Following dinner, Pres. R. F. Taylor presided at a brief business meeting at which 23 new members were introduced and welcomed into the fellowship of the Society. J. H. Van Alsburg, Chicago, a member of the A.S.H.V.E. Committee on Research, was presented by President Taylor. A special meeting of the Chapter in Fort Worth on March 22 was announced with Mr. Van Alsburg as speaker on Comfort and Research.

A Nominating Committee composed of C. R. Gardner, R. W. Kotzebue, and R. E. Chrone was appointed to report to the April meeting a list of nominees for Chapter officers for the year 1938-39.

Prof. F. E. Giesecke, 2nd vice-president of the Society, presented President Gurney, whose discussion of the manner in which the HEATING, VENTILATING, AIR CONDITIONING GUIDE 1938 was compiled, the Society's research, and the results of the meeting of Chapter delegates at the 44th Annual Meeting of the A.S.H.V.E. in New York was exceptionally interesting and well received. After a short round-table discussion of Mr. Gurney's talk, the meeting adjourned, according to the report of Secy. W. H. Badgett.

Philadelphia Chapter

March 10, 1938. The regular meeting of the Philadelphia Chapter was held at the Edward Bok Vocational School, where dinner was served in the cafeteria, following which a short talk was given in behalf of the United Campaign.

In order to conserve time, Secy. C. B. Eastman reports that the reading of the minutes of the previous meeting was dispensed with. A report was given by the treasurer, and Vice-Pres. H. H. Erickson introduced H. F. Rettew, who called upon various men connected with the Board of Education who had contributed in the planning and erection of the Vocational School.

It was moved and seconded that a letter of thanks be written by the secretary to the Board of Education expressing the appreciation of the Chapter for the courtesy shown them.

The meeting was then divided into groups who were shown through the building. The inspection which took nearly two hours was confined largely to the mechanical equipment.

Oklahoma Chapter

March 28, 1938. A dinner meeting was held at the Oklahoma City Golf and Country Club with 31 members and guests of Oklahoma Chapter attending.

Following dinner, Pres. E. F. Dawson introduced the out-of-town guests and Charles Bowes, of the Daily Oklahoman. He then presented J. H. Van Alsburg, Hart and Cooley Mfg. Co., Chicago, Ill., who spoke on Gaseous Engineering. Mr. Van Alsburg, who is a member of the Society's Committee on Research, presented this subject in an interesting manner and stressed in particular the amount of research that has been done by the A. S. H. V. E. and the necessity of a great deal more in the future.

C. Z. Woodworth, consulting engineer, gave a brief description of the air conditioning system in a new residence, after which the members made a tour of inspection and heard a description of the various features of the structure.

Michigan Chapter

March 14, 1938. The regular meeting of the Michigan Chapter was held at the Wardell Hotel at 6:30 p. m. with approximately 65 members and guests present.

After dinner Pres. F. J. Feely asked for roll call and the reports of the secretary, treasurer, and the chairman of the Program Committee. W. G. Boales read President Feely's report on the Annual Meeting of the Society held in New York.

Prof. L. G. Miller, Michigan State College, discussed the annual short course sponsored by the *National Warm Air Heating and Air Conditioning Association* to be held at East Lansing, March 28-31, and invited Michigan Chapter to attend the banquet on March 30 with J. F. McIntire, 1st vice-president of the A. S. H. V. E., as speaker on Research and Air Conditioning—Magic Words.

President Feely then introduced T. W. Weigele, house heating engineer, Detroit City Gas Co., as the speaker of the evening. In discussing Providing Detroit with Natural Gas, Mr. Weigele pointed out some interesting problems which were encountered and solved regarding rates, pipe lines, pumping stations, pressures, and the method employed in changing all gas-burning appliances in Detroit from manufactured to natural gas.

According to the report of Secy. G. H. Tuttle the meeting adjourned at 10:30 p. m. after a rising vote of thanks to the speaker.

Iowa-Nebraska Chapter

March 8, 1938. A meeting of the Iowa-Nebraska Chapter was held at the Hotel Cornhusker, Lincoln, and was attended by 53 members and guests at dinner, while more than 100 were present for the technical program. A. L. Walters presided.

The minutes of the previous meeting were read and approved and reports were made for the following committees: Membership Committee—Henry Kleinkauf; Publicity Committee—W. J. Hennessy; and by Treas. F. L. D. Banner.

Prof. R. A. Norman, Iowa State College, offered a few comments pertaining to the procedure to be followed at the Heating, Ventilating and Air Conditioning Conference during April 6-8 at Ames, Iowa.

Secy. W. R. White read a letter from M. J. Stevenson, in which he tendered his resignation as president due to his new location in Chicago.

A brief report of the Board of Governors' meeting held previous to the dinner was given by Secretary White and the appointment of Vice-President Walters as president was announced. It was also decided that those whose dues were paid as of March 8 would be considered charter members of the Chapter.

Mr. Banner reported that the Chapter has a total of 38 members on its rolls.

It was moved that the Secretary write a letter to Mr. Stevenson thanking him for his part in initiating the formation of the Chapter and expressing regret at his leaving the local organization.

N. B. Delavan, chairman of the Meetings Committee, introduced the speaker of the evening, R. F. Rosenbach, Sidles Co., who gave a detailed discussion of an air conditioning installation in the Union Pacific Building, Omaha. He explained the procedure followed in making the preliminary survey, in deciding upon the type of installation to be used, and in selecting the equipment. He discussed the steps that were followed during the construction period, and explained the method used in placing the major equipment, particularly the large 9 x 10 ft air ducts attached to the outside of the building.

President Walters closed the meeting by thanking Mr. Rosenbach for his fine lecture and expressed the pleasure of the Chapter in having so many guests present.

Ontario Chapter

April 4, 1938. The meeting of the Ontario Chapter was held in the Royal York Hotel, Toronto, with 68 members and guests present, and was preceded by dinner.

H. H. Angus reported on the progress of the Code Committee.

W. R. Blackhall stated that at the next meeting of the Chapter the holder of the Ontario Chapter award to the University of Toronto student writing the best thesis on Air Conditioning would be announced.

The principal speaker of the evening was D. J. Purinton, Mahoney-Troast Construction Co., New York, who was introduced by E. R. Gauley. Mr. Purinton spoke on Radiant Heating. Considerable discussion followed and Prof. E. A. Allcut extended a vote of thanks to the speaker for his instructive talk.

Before the meeting adjourned, Pres. G. A. Playfair spoke of the death of T. K. Dobie, a member of Ontario Chapter.

March 7, 1938. When the regular March meeting of Ontario Chapter was held in the Royal York Hotel 67 members and guests were present and dinner was served at 6:30 p. m.

A. J. Dickey, chairman of the Nominating Committee, announced the following nominations for officers for the ensuing year:

President—H. B. Jenney
Vice-President—H. D. Henion
Secretary-Treasurer—H. R. Roth
Board of Governors—F. E. Ellis, C. Tasker, J. W. O'Neill.

As past president of the Chapter G. A. Playfair will automatically serve on the Board of Governors for the next year.

H. H. Angus reported on the Code Committee and stated that satisfactory progress was being made.

Mr. Tasker introduced the principal speaker for the evening, A. L. Waines, Johns-Manviile Co., who presented a talkie film entitled Heat.

Following this presentation a vote of thanks was extended to Mr. Waines by M. W. Shears.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 35 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn, the Council, urge the members to assume their share of responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by May 16, 1938, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
ALBRIGHT, C. B., Chief Air Cond. Engr., Lightning Elec. Service Co., Caldwell, N. J.	G. L. Larson John James	D. W. Nelson Webster Tallmadge
BROKAW, G. K., Student, Univ. of Calif., Oakland, Calif.	W. E. Leland B. F. Raber	V. H. Cherry L. M. K. Boelter (*Non-Member*)
BROWN, M. D., Mech. Engr., Northup & O'Brien, Archts., Winston-Salem, N. C. (*Advancement*)	L. W. Biebigheiser (*Non-Member*) G. R. Stinson (*Non-Member*)	W. R. James (*Non-Member*) G. E. Brewer (*Non-Member*)

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
CARPENTER, R. D., Student, Calif. Polytechnic, San Luis Obispo, Calif.	R. R. Mason (*Non-Member*)	C. E. Knott (*Non-Member*)
	N. Sharpe (*Non-Member*)	H. C. Figge (*Non-Member*)
COLLE, S. S., Owner, Air Cond. Engrg. Co., Montreal, Que., Can.	C. W. Johnson	H. G. S. Murray
	Leonard Martin	F. G. Phipps
DEGLER, H. E., Prof. Mech. Engrg., Univ. of Texas, Austin, Texas.	F. E. Giesecke	W. H. Severns
	R. F. Taylor	A. P. Kratz
DOWNING, C. B., Secy.-Treas. & Mgr., N. B. Downing Co., Milford, Dela.	F. I. Ponsell	C. Muirhead (*Non-Member*)
	M. C. Barnum	Grant Windsor (*Non-Member*)
EGGLESTON, H. L., Mgr., Natural Gas & Ref. Depts., Gilmore Oil Co., Los Angeles, Calif.	R. E. Phillips	Leo Hungerford
	W. S. Kilpatrick	H. B. Lauer
EVERETTS, JOHN, JR., Secy.-Treas. & Engr., Air & Refrigeration Corp., New York, N. Y. (*Advancement*)	W. L. Fleisher	J. L. Blackshaw
	W. E. Heibel	John James
FARROW, E. E., Pres. & Gen. Mgr., E. E. Farrow, Inc., Dallas, Texas.	R. F. Taylor	F. E. Giesecke
	R. K. Werner	H. W. Skinner
GARNEAU, LEO, Sales Engr., C. A., Dunham Co., Ltd., Montreal, Que., Can. (*Advancement*)	F. A. Hamlet	F. G. Phipps
	G. L. Wiggs	C. W. Johnson
GENRE, E. J., Sales, Norge Htg. & Air Cond., Phoenix, Ariz.	E. H. Kendall	P. M. Tidmarsh
	H. B. Lauer	B. U. Nowaish (*Non-Member*)
GORDON, C. W., Supt., A. G. Baird, Toronto, Ont., Can.	R. P. Allsop	H. R. Roth
	E. L. Bowerman	W. P. Boddington
GORDON, H. H. W., Resident Engr., Carrier Engrg. South Africa, Cape Town, South Africa.	Jacob Ehlers	C. A. von Christierson
	E. G. Carrier	D. C. Lindsay
GOSS, M. H., Partner, M. H. Goss Co., Detroit, Mich. (*Reinstatement*)	W. G. Boales	G. H. Tuttle
	R. K. Milward	G. D. Winans
GROSSENBACHER, H. E., Owner & Mgr., Grossenbacher Steel Furnace & Mfg. Co., St. Louis, Mo.	G. W. F. Myers	C. F. Boester
	E. E. Carlson	R. J. Tenkonohy
HOCHMAN, EUGENE, Student, Mass. Inst. of Tech., Cambridge, Mass.	James Holt	G. B. Wilkes
	H. C. Moore	Carl Svenson (*A. S. R. E.*)
JAKOFSKY, F. A., Student, New York Univ., New York, N. Y.	Henry Gitterman	W. T. Wrightson
	C. S. Pabst	E. E. Adams
LIPSCOMBE, H. W. J., Managing Director, The Lipscombe Air Cond. Co., Ltd., London, S. W. 1, England.	J. K. M. Pryke	W. G. Case
	Victor Kraminsky	Thomas Chester
LYNN, F. E., Chief Engr. Air Cond., Electric Products Corp., Pittsburgh, Pa.	J. J. La Salvia	M. L. Carr
	H. A. Canon	F. C. McIntosh
MAY, J. W., Instructor, Univ. of Kentucky, Lexington, Ky. (*Advancement*)	L. S. O'Bannon	E. A. Bureau (*A.I.E.E.*)
	Perry West	C. C. Jett (*A.S.M.E.*)
PENNEY, G. W., Mgr., Electro-Physics Div., Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa.	F. C. Houghten	T. F. Rockwell
	J. F. Collins, Jr.	E. C. Smyers
REED, W. H., 3rd, Sales Engr., Dravo Corp., Pittsburgh, Pa.	H. A. Canon	E. C. Smyers
	C. E. Parks	P. A. Edwards
SCALINGI, C. R., Student, Mass. Inst. of Tech., Cambridge, Mass.	James Holt	G. B. Wilkes
	H. C. Moore	Carl Svenson (*A.S.R.E.*)
SESSLER, R. E., Student, Mass. Inst. of Tech., Cambridge, Mass.	James Holt	G. B. Wilkes
	H. C. Moore	Carl Svenson (*A.S.R.E.*)
SHAW, J. A., Gen. Elec. Engr., Canadian Pacific Ry. Co., Montreal, Que., Can.	G. L. Wiggs	Leo Garneau
	C. W. Johnson	L. H. Laffoley
SIMONS, E. W., Design Engr., Redwood Mfrs. Co., San Francisco, Calif.	E. O. Kaup	G. J. Cummings
	R. A. Folsom	G. M. Simonson
SPRING, C. L., Htg. Engr., Des Moines Stove Repair Co., Des Moines, Iowa.	L. G. Miller	T. R. Johnson
	R. A. Norman	F. L. D. Banner
STEENKAMP, WILLEM, Student, Mass. Inst. of Tech., Cambridge, Mass.	James Holt	C. M. F. Peterson
	H. C. Moore	W. H. Jones (*A.S.R.E.*)
SUPON, DANIEL, Student, Carnegie Inst. of Tech., Pittsburgh, Pa.	C. M. Humphreys	F. C. McIntosh
	T. F. Rockwell	R. B. Stanger
TEVES, H. L., Gen. Mgr., N. V. Becht & Dyserinck, Amsterdam, Netherlands.	Thomas McDonald	M. M. Smith (*Non-Member*)
	D. Goedkoop Dzn. (*Non-Member*)	H. H. Wex (*Non-Member*)
TUTHILL, A. F., Student, Carnegie Inst. of Tech., Pittsburgh, Pa.	C. M. Humphreys	F. C. McIntosh
	T. F. Rockwell	R. B. Stanger
VLADIKOFF, E. I., Student, New York Univ., New York, N. Y.	R. D. Lambert	M. C. Giannini
	A. A. Adler	P. B. Gordon
YUSKA, L. J., Student, State Univ. of Iowa, Iowa City, Iowa.	H. O. Croft (*A.S.M.E.*)	T. C. Caywood (*Non-Member*)
	R. M. Barnes (*Non-Member*)	J. M. Russ (*Non-Member*)
ZUMWALT, ROSS, Member of Firm, Zumwalt & Darrah, Dallas, Texas.	L. S. Gilbert	C. R. Gardner
	H. J. Martyn	A. M. Harris

Candidates Elected

In the past issues of the JOURNAL of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Committee on Admission and Advancement and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. B-III, Sec. 8, of the By-Laws, the following list of candidates elected:

MEMBERS

BOND, H. H., Partner, Edward E. Ashley, Cons. Engr., New York, N. Y.

DAVIS, CHARLES, Chief Engr., Rathe Htg. Corp., New York, N. Y.

EKLUND, K. G., Consulting Engr., Stockholm, Sweden.

EVANS, B. L., Design Engr., Langenberg Htg. Co., St. Louis, Mo. (*Advancement*)

FAMILETTI, A. R., Chief Engrg. Draftsman, Navy Yard, Philadelphia, Pa. (*Advancement*)